Pulleys, Belts and Transmissions

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Waterwheels, Turbines, and Generators

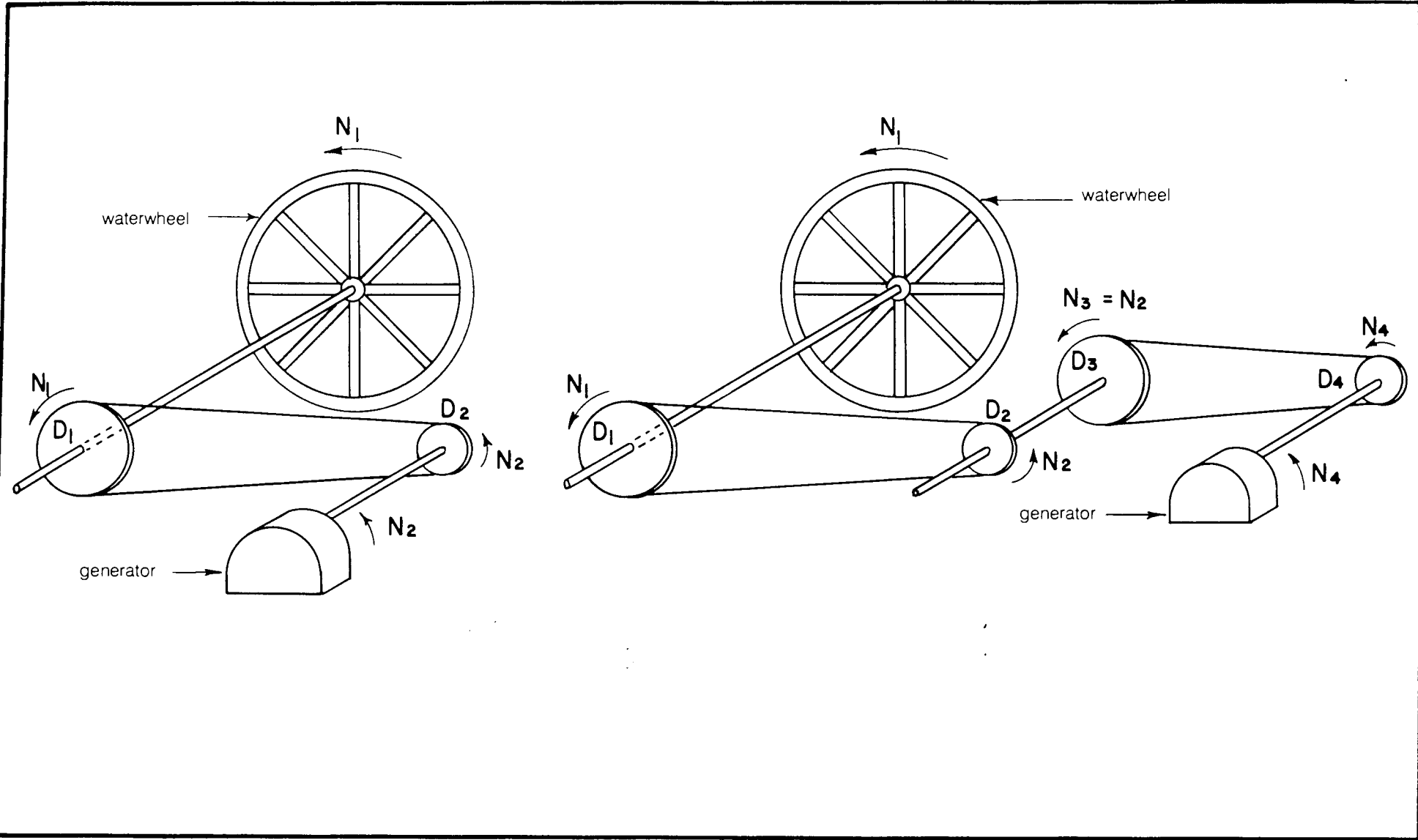
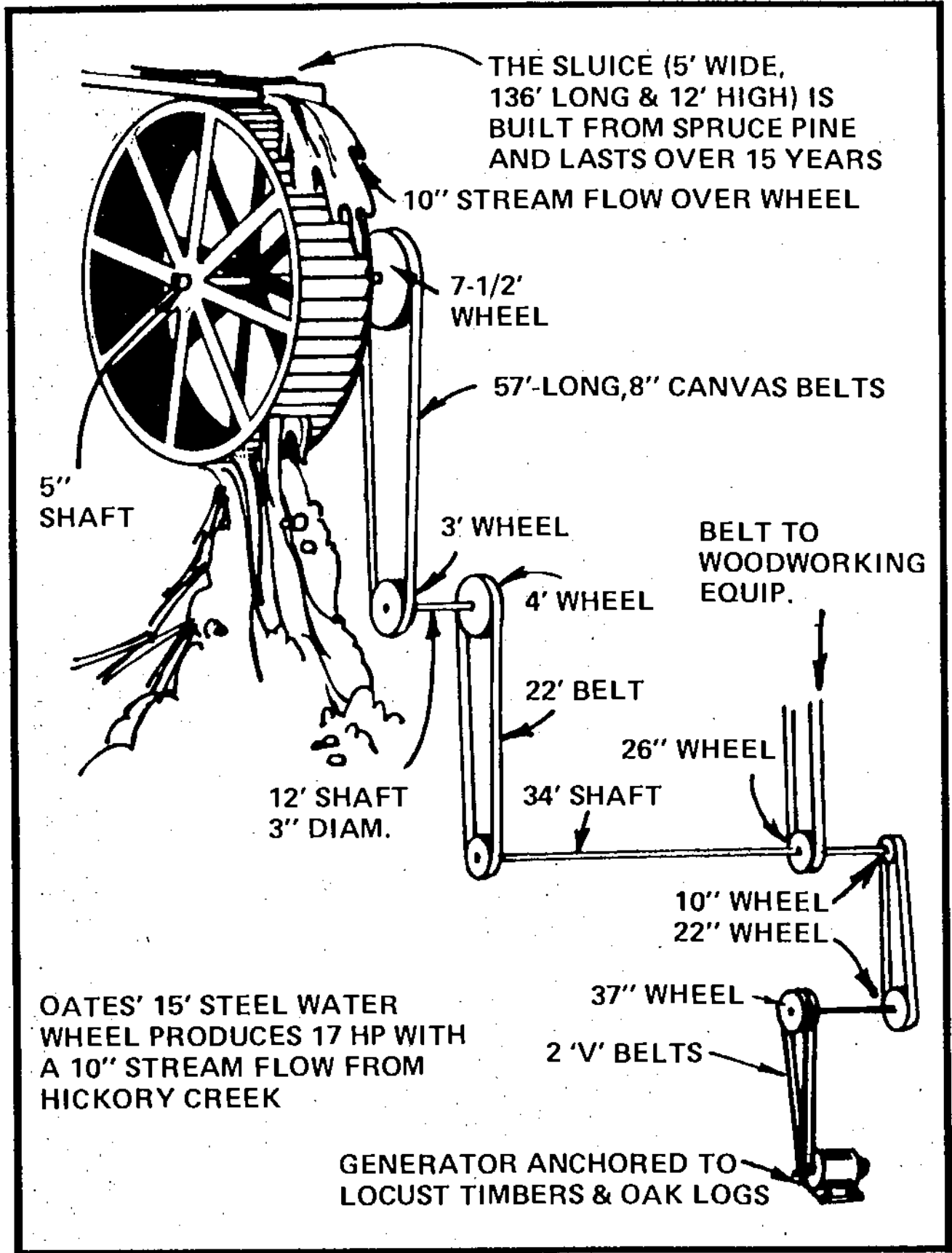
The energy in our stream of water is converted into mechanical, rotational energy by means of a waterwheel or a turbine. Waterwheels have the advantage of being rather "low technology"; they can be built without much in the way of special skills, materials, or tools. The shaft of a waterwheel can be coupled by means of belts and pulleys directly into such low-speed mechanical loads as saws, lathes, water pumps, and mills; but their low turning speed makes it difficult to couple them into electrical generators. A typical waterwheel turns at something like 5 to 15 rpm and, since an automobile generator needs 2000 to 3000 rpm to put out much current, a gear ratio of about 300:1 would be required. Such high gear ratios are quite difficult to attain, which severely limits the usefulness of the waterwheel–automobile-generator combination. There are generators, used in commercial wind-electric plants, which put out significant power at lower rpms; however they are considerably more expensive.

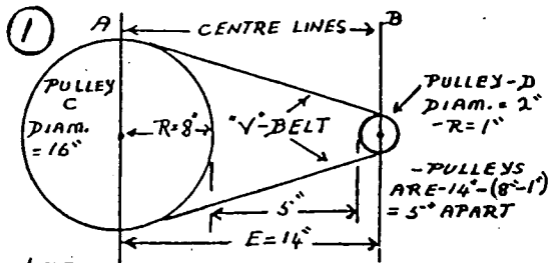
To speed up the rotation of a shaft, pulleys and belts can be used, as shown in Figure 3.29. The ratio of the speeds of the shafts is equal to the ratio of the diameters of the pulleys:

E. 3.15
$$\frac{N_2}{N_1} = \frac{D_1}{D_2}$$

To obtain a speed increase of, say, 20 to 1, the diameter of one pulley would have to be 20 times the diameter of the other. For large speed increases, the step-up should be done in stages, as shown in Figure 3.29. In this case, the total increase in speed is equal to the product of each pulley ratio:

E. 3.16
$$\frac{N_4}{N_1} = \frac{D_1}{D_2} \times \frac{D_3}{D_4}$$

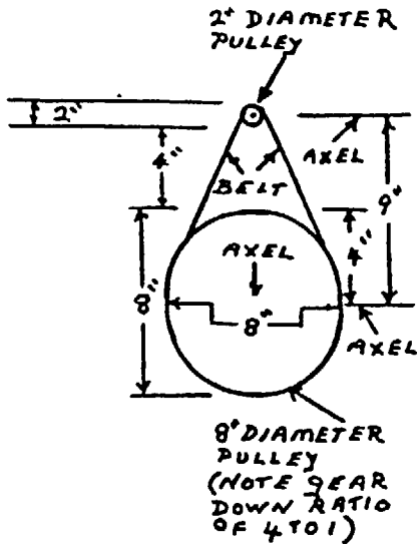
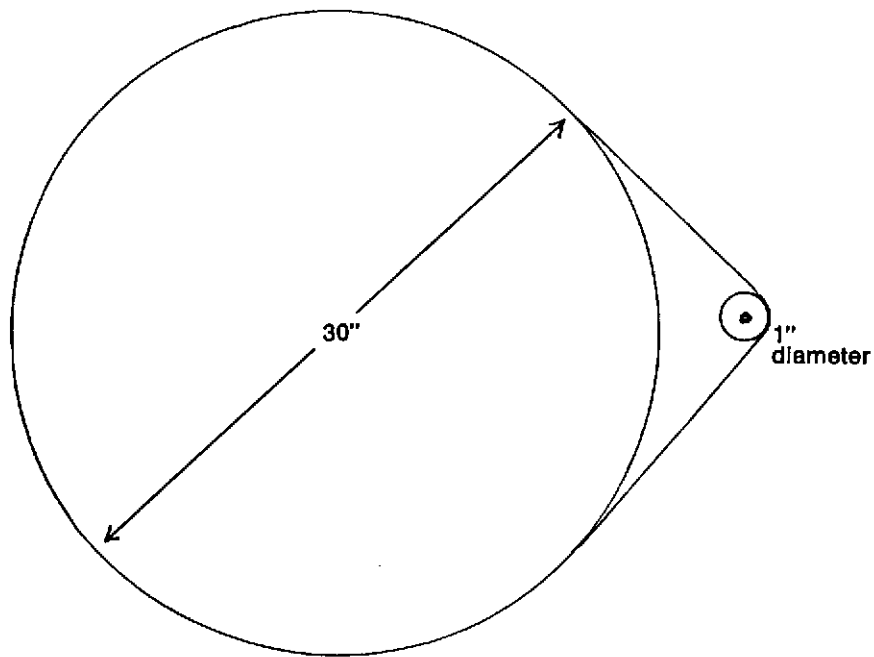
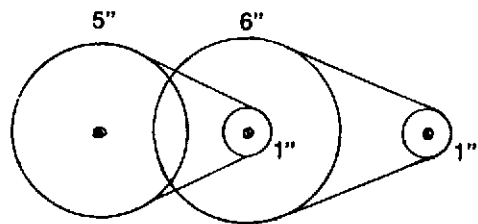
In spite of the difficulty, it is possible to use waterwheels for generating electricity: Thomas Oates, for example, has described a four-stage belt-drive system which he uses to step up a waterwheel rotating at 15 rpm to a generator rotating at 800 rpm (in *Mother Earth News A Handbook of Homemade Power*

Matching the slow speed of a waterwheel to the high-speed requirements of a generator by belts and pulleys.

Oates uses a series of canvas belts and nine drive pulleys to transfer power from his water wheel to a generator and other equipment.

GEAR-UP-V-BELT PRINCIPLES

NOTE
-V-BELTS TRANSMIT POWER THROUGH THEIR SIDE WALLS-BY WEDGING ACTION-
-PULLEYS MUST BE FAR ENOUGH APART TO ALLOW BELTS TO COVER HALF OF EACH PULLEY-

RULE- DISTANCE (E)-BETWEEN THE PULLEY CENTRES (A-AND-B)-MUST EQUAL THE DIFFERENCE BETWEEN THE TWO PULLEY DIAMETERS-
∴ DISTANCE (E) = 16" (C) LESS 2" (D) = 14"

BELT LENGTH = 2 × (E) + ½ OF CIRCUMFERENCE (C) + ½ OF CIRCUMFERENCE (D)-

NOTE-PLAIN LEATHER BELTS OR CHAINS CAN REPLACE "V"-BELTS-

2" DIAMETER
PULLEY
2"
4"
AXEL
BELT
9"
4"
AXEL
8"
8"
AXEL
8" DIAMETER
PULLEY
(NOTE GEAR
DOWN RATIO
OF 4 TO 1)

Pulley system

Pulleys and drive belts Check the alignment of drive pulleys by using a long straight edge or taut string (Figure 13.6). The string should lie flat across the face of each pulley if they are correctly aligned.

A loose drive belt will slip causing power loss, overheating and rapid wear of the belt. A tight drive belt will consume extra power, rapidly wear the belt and damage the pump or engine bearings. Therefore, ensure drive belts are adjusted to the manufacturer's recommended tension. Rule of thumb: there should be some give in the belt under firm finger pressure. Better a belt which is too loose (it wears itself out) than too tight (it may damage bearings).

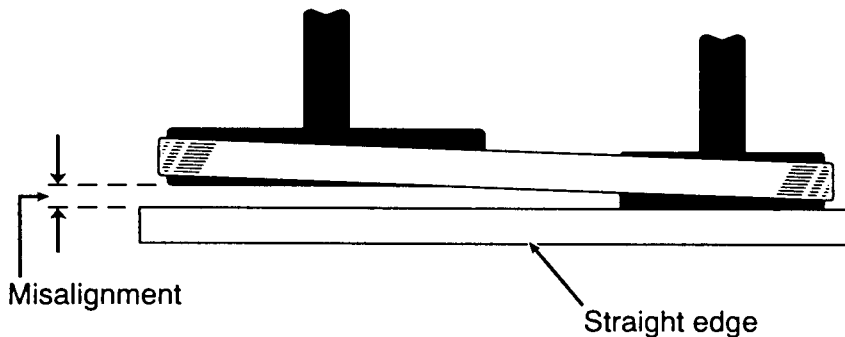



Figure 13.6 Checking the alignment of pulleys

How to Find the Size of a Pulley.— In order to run any farm machine at its proper speed, that is the speed which the manufacturer recommends as the best, you must be able to calculate the size of the pulley needed on the machine.

The pulley on your tractor is, of course, of a fixed size and if you want to run, say, an ensilage cutter at a given speed, a *ratio* between the sizes of the tractor pulley and the ensilage cutter pulley must be had. You can easily find the size needed by using the following formula:

$$\text{Diameter of Machine Pulley} = \frac{\text{Diam. of Tractor Pulley} \times \text{R.P.M. of Tractor Pulley}}{\text{R.P.M. of the Machine Pulley}}$$

Now let us take an example. An ensilage cutter, or other machine, is to be run at a speed of 800 r. p. m., by a tractor whose pulley is 16 inches in diameter and which is driven at a speed of 400 r. p. m. What must be the size of the pulley on the ensilage cutter or other machine?

Substituting now the known figures for the formula above we have

$$\text{Diameter of Machine Pulley} = \frac{16 \times 400}{800}$$

or worked out

$$\text{Diameter of Machine Pulley} = \frac{6400}{800} = 8 \text{ inches}$$

and 8 inches is the diameter of the pulley you want on your ensilage cutter or other machine.

How to Find the Belt Speed.— The belt speed of your tractor is the rate of travel of any one point on the belt and it is measured in feet per minute. You can find it by the following formula:

$$\text{Belt Speed} = \frac{(\text{Diameter of Tractor Pulley} \times 3.1416) \times \text{R.P.M. of Tractor Pulley}}{12 \text{ inches}}$$

where the diameter of the tractor pulley $\times$ 3.1416 = the circumference of the tractor pulley in inches.

As an example, suppose the diameter of the tractor pulley is 18 inches and its speed is 550 r. p. m. Substituting these figures in the formula we have,

$$\text{Belt Speed} = \frac{18 \times 3.1416 \times 550}{12}$$

or,

$$\text{Belt Speed} = \frac{31{,}200}{12} = 2{,}600 \text{ feet},$$

and 2,600 feet is the speed of the belt in feet per minute.

Rules for Pulley Sizes.—The following formula can be used to calculate sizes of pulleys for motor drives:

$$d = \frac{D \times N}{n} \qquad n = \frac{D \times N}{d}$$

D = Diameter of driver.
d = Diameter of driven pulley.
N = Revolutions per minute of driver.
n = Revolutions per minute of driven pulley.

Speed of Pulleys.—When the diameter of the driven pulley is given, to find its number of revolutions proceed as follows: Multiply the dia-

meter of the driver by the number of its revolutions, and divide the product by the diameter of the driven. The quotient will be the number of revolutions of the driven pulley.

When the diameter and revolutions of driver pulley are given, to find the diameter of the driven pulley that will make any given number of revolutions in the same time, proceed as follows: Multiply the diameter of the driver pulley by its number of revolutions, and divide the product by the number of revolutions of the driven pulley. The quotient will be the diameter of the driven pulley.

To ascertain the size of the driver pulley proceed as follows: Multiply the diameter of the driven pulley by the number of revolutions you wish to make, and divide the product by the revolutions of the driver pulley. The quotient will be the diameter of the driven pulley.

PULLEY SPEEDS. The principle applied to gearing in regard to the ratio between the speeds of two shafts may be directly applied to pulleys, with the only difference that the number of inches to the diameter of the pulley should be substituted for the number of teeth in the gear. (See GEAR TRAINS.)

Assume that a shaft is required to make 300 revolutions per minute, and that it is driven from a lineshaft making 180 revolutions per minute, as indicated in Fig. 1. The pulley on the lineshaft is 15 inches in diameter. What should the diameter of the pulley on the shaft making 300 revolutions per minute be made? As the belt on the two pulleys runs at the same speed as the periphery (circumference) of either of the pulleys, it is clear that the peripheries of both pulleys run at the same speed, providing there is no slip between the belt and the pulleys. The pulley running a smaller number of revolutions must be larger in order that its periphery may run at the same speed as the periphery of the pulley making a greater number of revolutions. The circumference of a circle (and, therefore, also the circumference of a pulley) equals the diameter × 3.1416. Therefore, the circumference of the pulley making 180 revolutions and having a diameter of 15 inches, passes, in one minute, through a distance equal to 180 times its circumference, or 180 × 15 × 3.1416.

The circumference of the pulley making 300 revolutions must pass through the same distance in one minute; therefore, for *each* revolution, this pulley must pass through the distance 180 × 15 × 3.1416 divided by 300. This, then, would equal the circumference of the smaller pulley; but the circumference also equals the *diameter* × 3.1416. Therefore:

$$\frac{180 \times 15 \times 3.1416}{300} = \text{diameter of smaller pulley} \times 3.1416.$$

As 3.1416 enters as a factor on both sides of the equals sign, it can be canceled. Then:

$$\frac{180 \times 15}{300} = \text{diameter of smaller pulley.}$$

From this, the following rule for the relation between the sizes of pulleys and the number of revolutions of two shafts can be formulated:

The number of revolutions of one shaft multiplied by the diameter of the pulley on the same shaft, divided by the number of revolutions of the second shaft, gives the diameter of the pulley on the second shaft.

This rule can be expressed as a formula:

$$\text{Diameter of pulley on second shaft} = \frac{\text{revolutions of first shaft} \times \text{diam. of pulley on first shaft}}{\text{revolutions of second shaft}}.$$

If one pulley makes 200 revolutions while another pulley makes 100 revolutions, the *speed ratio* between the two pulleys is 2 to 1. If one pulley makes 200 revolutions while another makes 50 revolutions, the speed ratio is 4 to 1, because one shaft makes four times as many revolutions as the other. If one shaft runs at 200 revolutions and another at 140 revolutions, the speed ratio would be 200 to 140. By canceling equal factors in 200 and 140,

reduce the ratio so as to express it with smaller numbers. In this case the ratio would be 10 to 7, because $\frac{200}{140} = \frac{20}{14} = \frac{10}{7}$.

If very accurate results are required, one thickness of the belt should be added to the diameter of the pulley itself, and the dimension thus obtained should be used in the preceding formulas instead of the diameter of the pulley rim. If the pulley is 5 inches in diameter and the belt $\frac{1}{8}$ inch thick, the diameter to be used in the formulas should be $5\frac{1}{8}$ inches. The results obtained in this manner will be very accurate provided there is no slipping of the belt on either of the pulleys. For ordinary practical purposes, however, it is customary to figure with the diameter of the pulley rim, taking no account of the thickness of the belt.

The following rules apply to the practical questions relating to pulley speeds and diameters that are most frequently met with in shop installations.

Speed of Driven Pulley Required. — Diameter and speed of driving pulley, and diameter of driven pulley are known. *Rule:* Multiply the diameter of the driving pulley by its speed in revolutions per minute, and divide the product by the diameter of the driven pulley.

Example: — If the diameter of the driving pulley is 15 inches and its speed, 180 revolutions per minute, and the

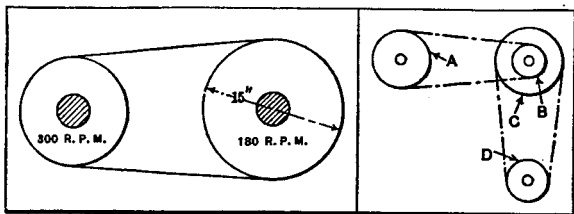



Fig. 1 Fig. 2

diameter of the driven pulley, 9 inches, then the speed of the driven pulley $= \frac{15 \times 180}{9} = 300$ revolutions per minute.

Diameter of Driven Pulley Required. — Diameter and speed of driving pulley, and revolutions per minute of driven pulley are known. *Rule:* Multiply the diameter of the driving pulley by its speed in revolutions per minute, and divide the product by the required speed of the driven pulley.

Example: — If the diameter of the driving pulley is 24 inches and its speed, 100 revolutions per minute, and the driven pulley is to rotate 600 revolutions per minute, then the diameter of the driven pulley $= \frac{24 \times 100}{600} = 4$ inches.

Diameter of Driving Pulley Required. — Diameter and speed of driven pulley, and speed of driving pulley are known. *Rule:* Multiply the diameter of the driven pulley by its speed in revolutions per minute, and divide the product by the speed of the driving pulley.

Example: — If the diameter of the driven pulley is 36 inches and its required speed, 150 revolutions per minute, and the speed of the driving pulley is 600 revolutions per minute, then the diameter of the driving pulley

$$= \frac{36 \times 150}{600} = 9 \text{ inches.}$$

Speed of Driving Pulley Required. — Diameters of driving and driven pulleys, and speed of driven pulley are known. *Rule:* Multiply the diameter of the driven pulley by its speed, and divide the product by the diameter of the driving pulley.

Example:—If the diameter of the driven pulley is 4 inches, its required speed, 800 revolutions per minute, and the diameter of the driver, 26 inches, then the required speed of the driver $= \frac{4 \times 800}{26} = 123$ revolutions per minute, approximately.

Speed of Driven Pulley in Compound Drive Required. — Diameters of pulleys A, B, C, and D (see Fig. 2), and speed of pulley A are known; find speed of pulley D. *Rule:* Divide product of diameters of driving pulleys by product of diameters of driven pulleys, and multiply quotient by speed of first driving pulley.

Example:—If the diameters of the driving pulleys A and C are 18 and 24 inches; the diameters of the driven pulleys B and D, 12 and 13 inches; and the speed of the driver A, 260 revolutions per minute; then the speed of the driven pulley D equals:

$$\frac{18 \times 24}{12 \times 13} \times 260 = 720 \text{ revolutions per minute.}$$

Pulley Diameters in Compound Drive Required. — Speeds of driving and driven pulleys are known; find diameters of the four pulleys A, B, C, and D. *Rule:* Place the speed of the driving pulley as the numerator of a fraction, and the speed of driven pulley as the denominator, and reduce this fraction to its lowest terms; then resolve both the numerator and denominator into two factors, and multiply each "pair" of factors (a pair being one factor in the numerator and one in the denominator) by a trial number which will give pulleys of suitable diameters.

Example: — If the speed of pulley A is 260 revolutions per minute, and the required speed of pulley D is 720 revolutions per minute, find the diameters of the four pulleys. The fraction $\frac{260}{720}$ reduced to its lowest terms is $\frac{13}{36}$, which represents the required speed ratio. Resolve $\frac{13}{36}$ into two factors; $\frac{13}{36} = \frac{1 \times 13}{2 \times 18}$. Multiply by trial numbers 12 and 1:

$$\frac{(1 \times 12) \times (13 \times 1)}{(2 \times 12) \times (18 \times 1)} = \frac{12 \times 13}{24 \times 18}$$

The values 12 and 13 in the numerator represent the diameters of the *driven* pulleys B and D, and values 24 and 18 in the denominator, the diameters of the *driving* pulleys.

Length of Belt on Pulleys. — A simple rule, which can be used with fair accuracy when the pulley diameters are nearly equal, is as follows: Add the diameters of the two pulleys, divide the sum by 2, and multiply the quotient by $3\frac{1}{7}$; then add to the product twice the distance between the centers of the shafts. Expressing this rule as a formula, in which D = diameter of large pulley; d = diameter of small pulley; C = center distance between shafts; L = belt length:

$$L = \frac{D + d}{2} \times 3\frac{1}{7} + 2\,C.$$

Example: — Diameter of large pulley equals 15 inches; diameter of small pulley equals 13 inches. The center distance is 92 inches. Find length of belt.

$$L = \frac{15 + 13}{2} \times 3\frac{1}{7} + 2 \times 92 = 228 \text{ inches.}$$

The following suggestions are based entirely on the excellent work recently done and published by Prof. P. la Cour in Denmark on behalf of that Government, which has in that particular placed itself ahead of other countries—considerably to the advantage of many of its villages and isolated dwellings. The reader must be prepared to experiment a little—not indeed in principles but in details of apparatus to suit his own case—but may rest absolutely assured that the method is quite practical and satisfactory.

There are two main difficulties in applying a power so variable and intermittent as wind to the production and supply of electricity. There must, first, be a means of automatically switching on the dynamo to a set of accumulators whenever the former is in a position to deliver current, the same apparatus cutting it out when the power falls away. Secondly, means must be adopted whereby an increase of wind-power beyond the normal amount required to just work the dynamo shall not affect the output by increasing either voltage or current. Both these ends have been attained by La Cour with the simplest apparatus imaginable.

A consideration of the second question raised will show why it is necessary to decide on a definite wind-velocity as being that at which any given windmill shall supply its "normal" output. By rating it low, say a wind of 9 miles per hour, it is possible to keep a dynamo working nearly every day in the year and for twelve hours out of the twenty-four. But the power of the wind at 9 miles an hour is only a quarter of that at 15 miles an hour, and although the latter only blows about half the total number of days in a year, and then for only about nine or ten hours a day, its total output is greater than the other. Another point to be considered is that a very small dynamo is much less efficient, so that a double loss is experienced if too much constancy of work is aimed at. Of course, in a large installation these points have less emphasis, and it becomes desirable to run the plant at a lower wind-rating (in other words, use a comparatively large mill), the only limiting factor being the initial cost of the plant.

In a wind-driven generating plant the following points should be noted. The windmill itself should be self-regulating (as, for example, that described in Chap. V.), and fitted with tail so as to turn to face all possible winds. The dynamo should be shunt-wound, so that an increase in the external resistance tends to raise the terminal voltage. If necessary, this tendency may be increased by having one or two resistance coils in series with the shunt-winding, these coils being automatically cut out as the external resistance rises and current falls. A low-speed machine is certainly preferable, the speed of a windmill being rather low itself. The accumulator is a vital point: it should have a large capacity, as on this depends its ability to maintain a supply over a longer period of calm; yet as it is undesirable for any accumulator to remain long at a low state of charging, care must be taken to avoid draining it—especially if a spell of calm weather seems likely.

The whole of the electrical apparatus is shown diagrammatically in fig. 73, the only part needing much description being the automatic switch, further illustrated in three views in fig. 74. This consists of two electro-magnets, EM, each like an ordinary bell-magnet, and wound with fine wire, but with an extra winding of a few turns of thick wire, exactly like a compound-wound dynamo field magnet. A horse-shoe permanent magnet, PM, is suspended so that its poles lie opposite and near to the poles of the electro-magnets, and swings by means of the pivot screws which work in a *brass* (or non-magnetic) block, B. This block also carries the copper rod CR, each end of which turns downward into the wooden cups 1 and 2, containing mercury, matters being so arranged, however, that the end 1 is always in the mercury whichever way PM is swung, while 2 only touches the mercury when that end of CR is drawn downwards.

The switchboards present no special features. By following out the connections it will be seen that any agreed number of cells can be switched on to the dynamo, while any independent number can be caused to supply the lamps. This latter arrangement is desirable to allow for drop of voltage during discharge, also to provide for losses in mains and for an extra cell or two in case of accident to others.

The action of the automatic switch is as follows: Assuming the dynamo to be still, or running at too low a speed to furnish current, it will be seen that the battery is energising the electro-magnets EM through the fine wire-coils, the current passing also through the armature of the dynamo. The winding of EM is such that the current in this direction attracts the poles of PM to the right and so raises the end, 2, of CR out of the mercury. Only a very small current is required, or allowed, to be thus wasted. Supposing now the wind to increase sufficiently to raise the speed of dynamo so much as to be able to supply current, the first effect will be to reduce the current in EM to nil and then to reverse it, altering the polarity of the electro-magnets and throwing the lower end of magnet PM over to the right. This, by dipping the end 2 of CR into the mercury, makes connection between the dynamo and accumulator, the charging of which at once begins. The effect of the thick-wire coils on EM is to hold the magnet switch more securely during charging. The opposite action—that of throwing out the dynamo when the speed fails—is obvious on inspection.

There would be twelve accumulator cells, each of from 150 to 200 ampere-hour capacity, which would be easily capable of dealing with the full current for twenty-four hours' continuous charging. The capacity mentioned is the maximum suitable for the given plant, but the minimum may be anything down to twelve pocket-batteries, if so desired. Within the limits given, the greater the capacity the more the independence of conditions of wind.

With regard to the automatic switch, a little experimenting and adjusting will be needed to ensure its correct working. The electro-magnets may be two ordinary bell-magnets, wound with No. 36 wire, the bobbins being about 1¼ inches long and 1 inch diameter outside. A resistance may be needed in series with this winding, or the effect may be tried of connecting up only six of the cells to these coils, the six on the left-hand side in fig. 73 being, of course, selected. All four bobbins will be joined in series. Over the fine wire on each bobbin will be wound from six to twelve turns (to be determined by experiment) of No. 16 or 14 gauge cotton-covered wire, the winding being in same direction as the fine wire in each case, so that the current is a reinforcing one when being supplied from the dynamo. The balance of the permanent magnet can be adjusted by moving the copper rod CR either to right or left.

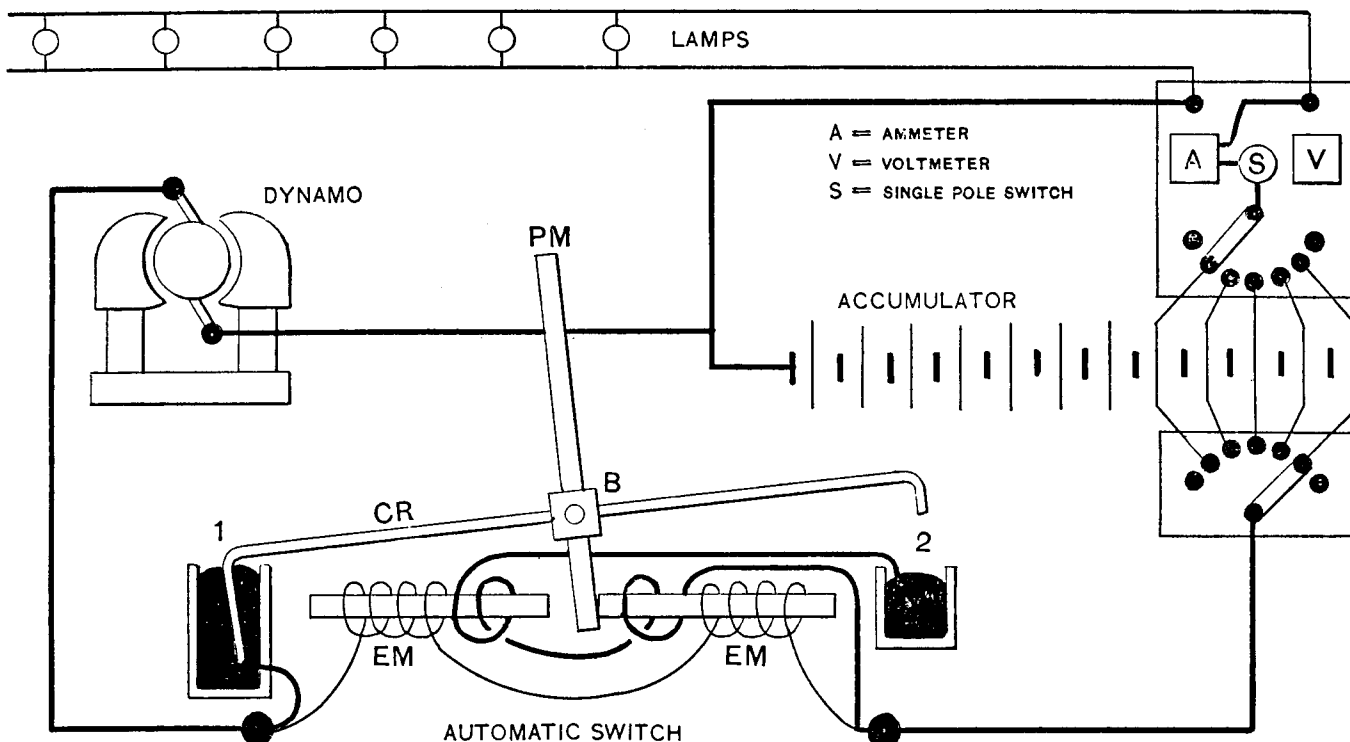



Fig. 73.— Diagram of Connections for a Wind-driven Electrical Installation.

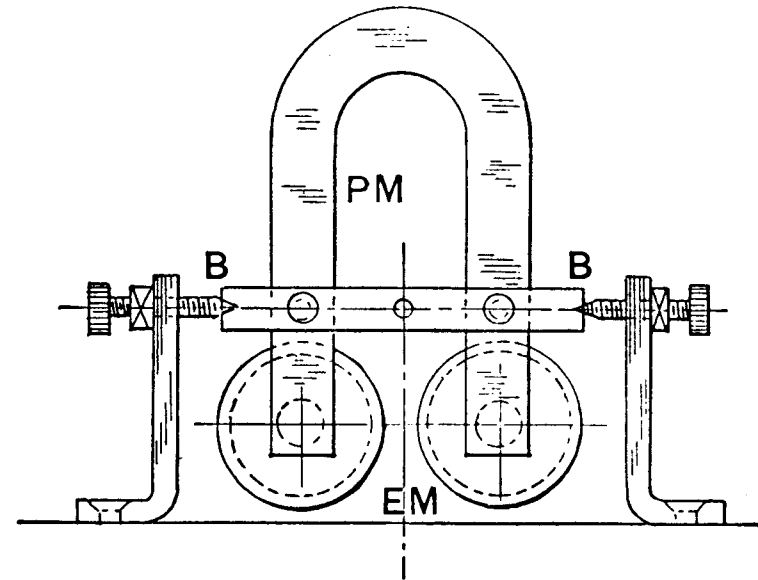
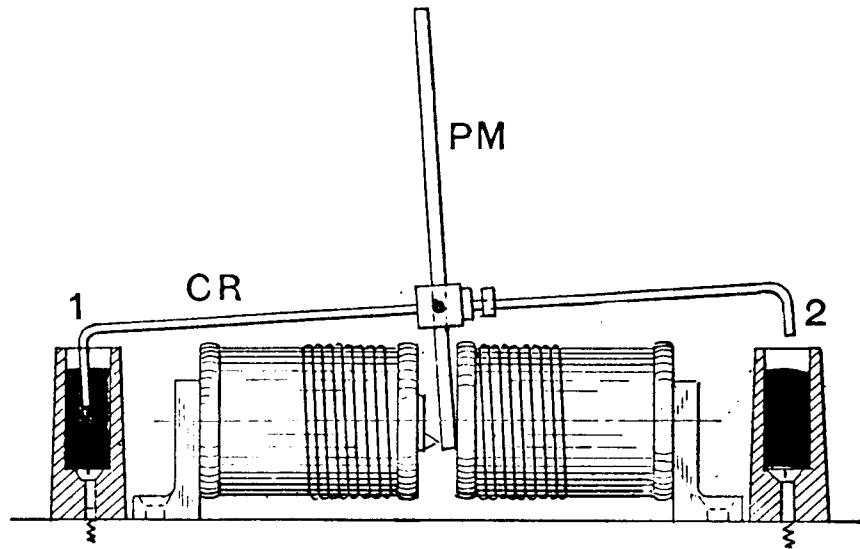
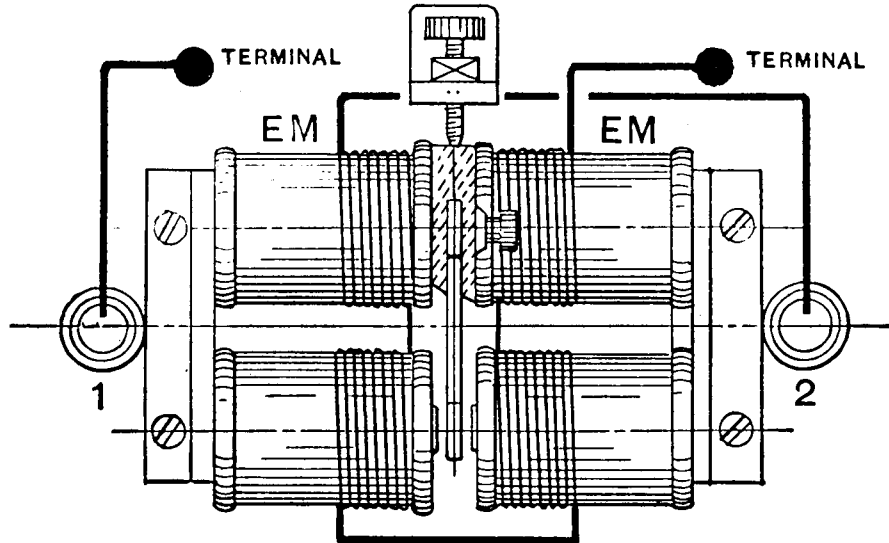



Fig. 74.—Automatic Switch for Wind Electrical Plant.

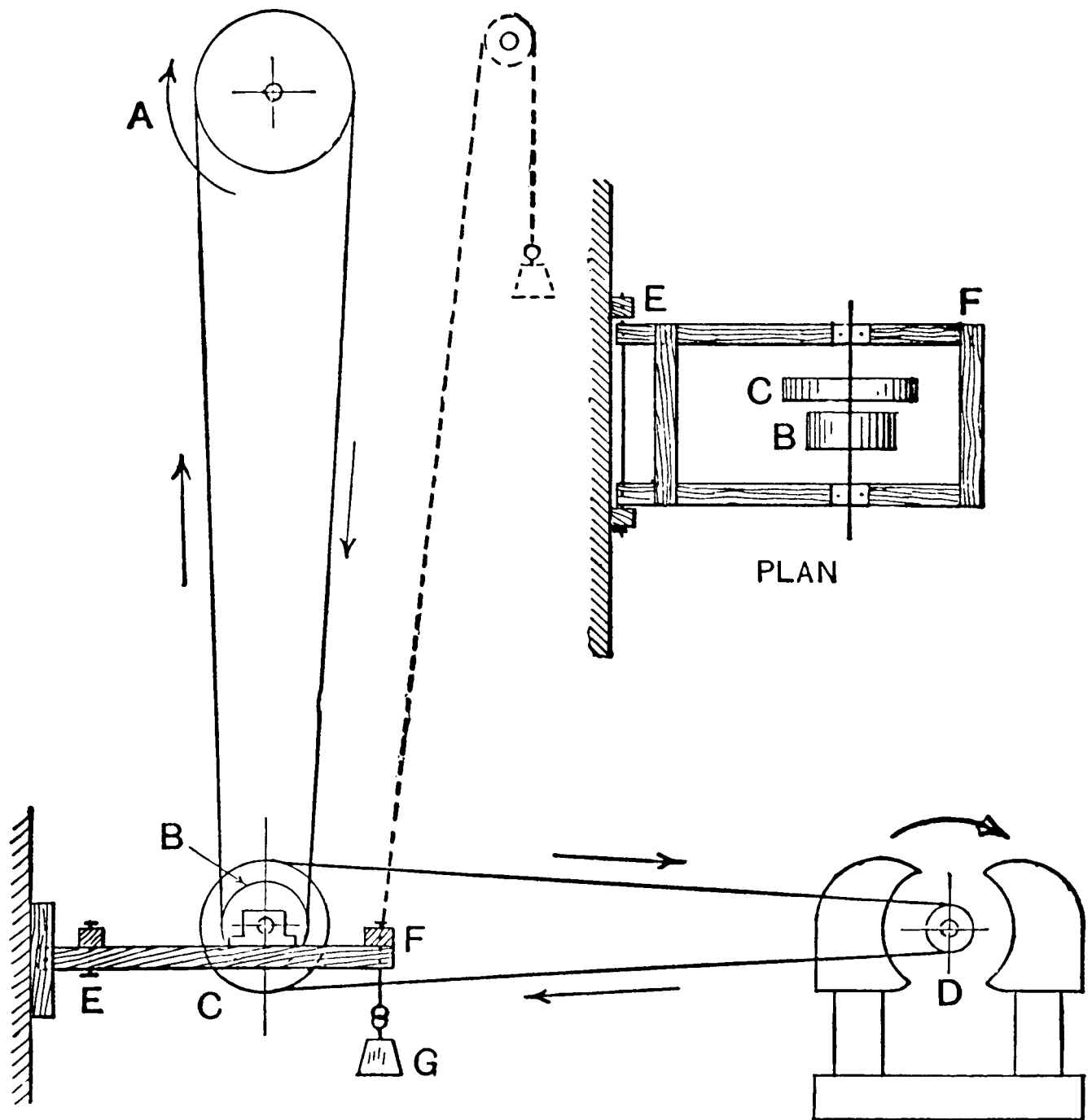



FIG. 75.—Driving Belt Arrangement for Wind Electrical Plant.

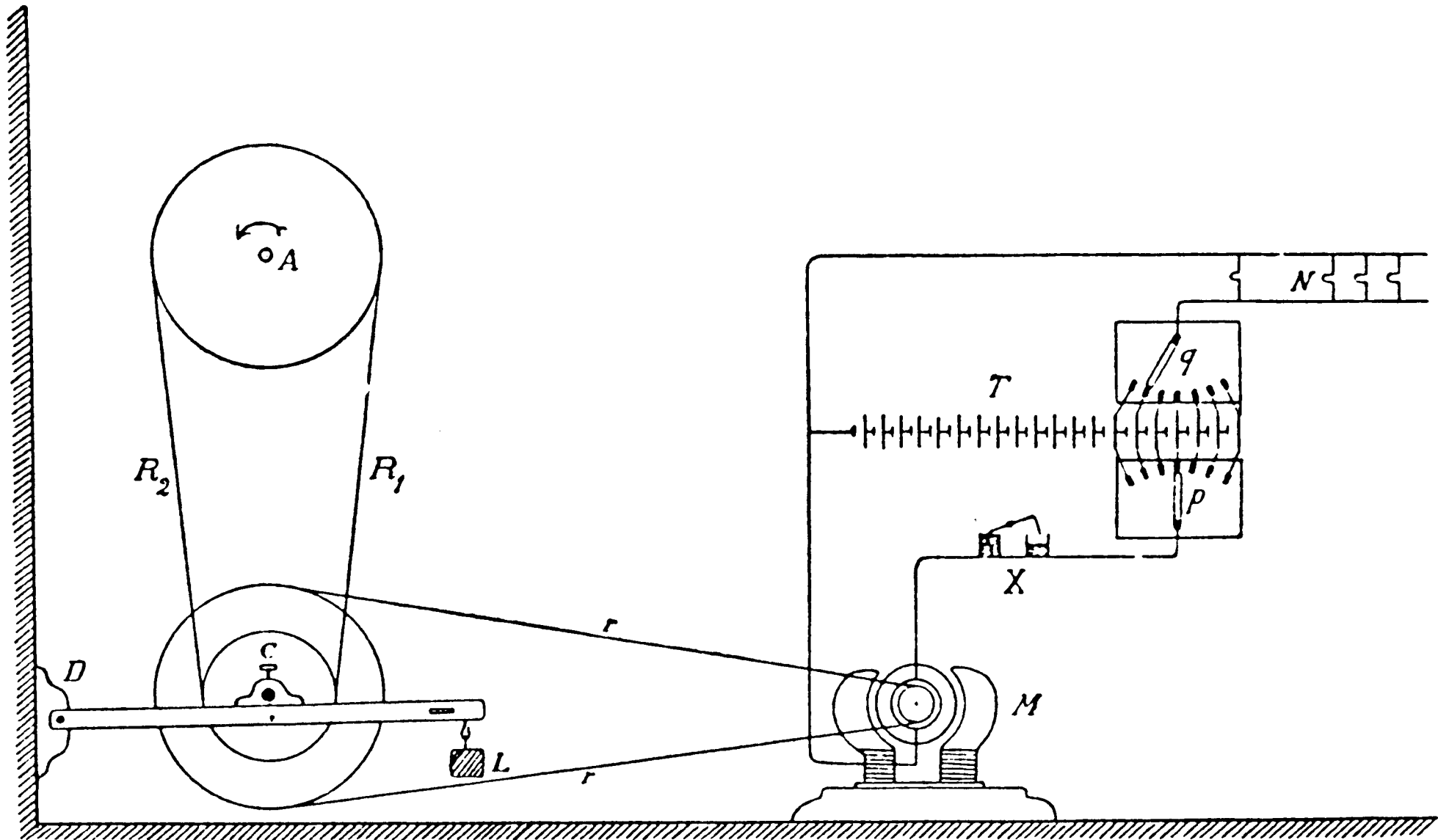



Fig. 5. Konstant elektrisk Strøm.

The apparatus required to maintain the dynamo at the right speed when that of the mill itself ranges too high is a system of belts and pulleys, shown in fig. 75. Here A is an ordinary pulley with the usual curved face; B, a rather wide, flat-surfaced pulley; C and D, again, ordinary pulleys; C and B being fast on one shaft. This shaft is carried on the light timber frame EF, hinged at E, and carrying a weight G at the other end.

It will be seen that this arrangement provides for a constant pull on the belt between A and B. It may be found that this pull is too great even without the weight G, in which case a cord (shown dotted) takes its place, and, by means of a pulley overhead and another weight, takes off some of the load.

The belt CD has no special feature beyond being thin, supple, and even. That between A and B, however, must be specially smooth on its running surface, and must in addition be thoroughly well oiled. On this depends the peculiar result to be obtained. It is found that when the weight G has been properly adjusted, and other details of current supply, etc., decided upon by experiment, no matter how much faster than normal A is compelled to run by the wind, the speed of B remains constant or with just sufficient variation to meet the slightly varying conditions required by the dynamo, the belt slipping on B at the higher speeds. The principle, of course, is not new; but its application in the present instance, together with the automatic switch, is an excellent example of mechanical adaptation.

The details of the whole of the apparatus must necessarily be worked out by individual requirements: the following suggestions, however, are added as an example, the instance chosen being the 10-foot windmill described in the last chapter. This windmill, working in a 15- or 16-mile breeze, should have an output of about ¼ H.P. Allowing for losses in dynamo, gearing, and belts, it may be assumed that a dynamo of 100-watts output would be the right machine for the available power. The voltage chosen might well be 25, this being its lowest rate at normal speed, which may be assumed at 1500 revs. per minute.

Under these circumstances, and assuming pulley A (fig. 75) to run at 200 revs. per minute (by whatever gearing used), A might be 12 inches diameter × 2 inches width; B, 6 inches × 3; C, 8 inches × 2; and D, the dynamo pulley, 2 inches × 2 inches. This gives a rather higher ratio than is required—an error on the right side. The belt between A and B should be 1½ inches × 3/16 inch, the pulleys being about 6 feet centres, and belt CD 1 inch wide × ⅛ inch thick, also with about 6 feet drive.

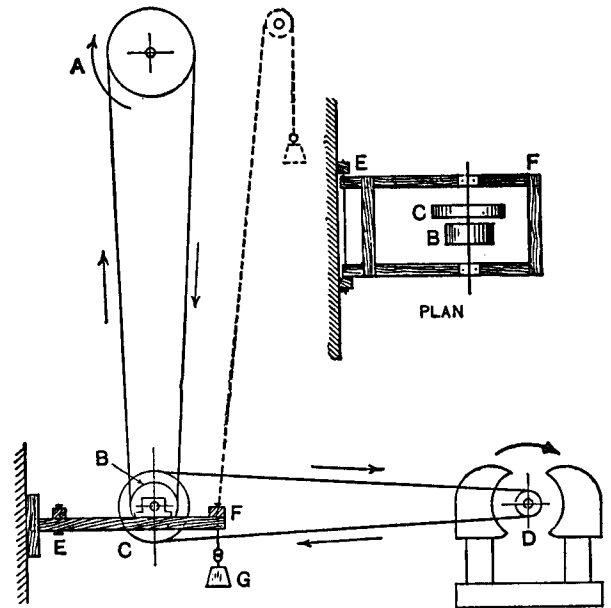



FIG. 75.—Driving Belt Arrangement for Wind Electrical Plant.

Belt Drives: Fundamental Concepts

If the driving shaft is at a considerable distance (up to 15 m) from the driven shaft, the moment of forces can be transmitted by a flexible connector using the frictional forces developed between the surface of the pulley and the flexible body.

Fig. 112.

Belts and ropes are used as a flexible connector for this kind of drives (Fig. 112*a* through *d*).

The peripheral speed on the driving pulley (Fig. 112*a*) is

$$v_1 = \frac{\pi D_1 n_1}{60}.$$

The peripheral speed on the driven pulley is

$$v_2 = \frac{\pi D_2 n_2}{60}.$$

If no account is taken of slipping between the belt and the pulley, the speeds v_1 and v_2 must be equal

$$v_1 = v_2$$

and

$$\frac{\pi D_1 n_1}{60} = \frac{\pi D_2 n_2}{60}.$$

Cancelling, we obtain

$$D_1 n_1 = D_2 n_2 \quad \text{or} \quad \frac{n_1}{n_2} = \frac{D_2}{D_1}.$$

In practice $v_1 \neq v_2$ since some elastic slippage always occurs between the belt and the pulley rim. Usually 1 to 3 per cent of the peripheral speed is lost in slipping. Introducing this correction, we obtain

$$v_2 = (1 - \varepsilon)\, v_1,$$

where ε is the slip ratio ($\varepsilon = 0.01$ to 0.03). Then

$$i_{1-2} = \frac{n_1 (1 - \varepsilon)}{n_2} = \frac{D_2}{D_1}. \qquad (127)$$

Elastic slippage is unavoidable in the operation of a belt drive under loading. This is not to be confused with parasitic slippage due to overloading or poor design.

§ 14. BELT DRIVES

Round, flat, and V-belts are used in machine tool drives. The directions of rotation of the pulleys mounted on the driving and driven shafts *coincide for open belts* (Fig. 30 *a* and *b*) *and are opposite for crossed belts* (Fig. 30 *c*). The speed of the shaft, on which the

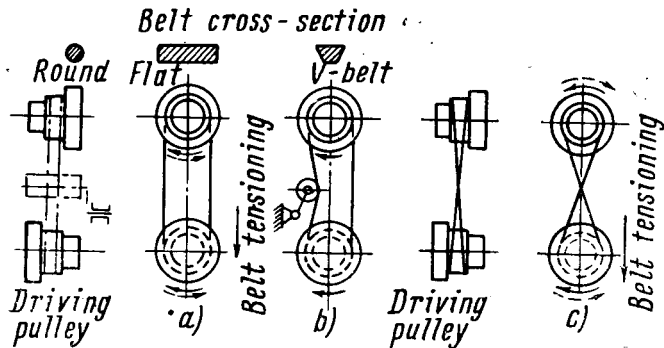



Fig. 30. Belt drives.
a and *b*—open belts, *c*—crossed belt.

smaller pulley is rigidly mounted, is higher than that of the shaft on which the larger pulley is rigidly mounted. To increase the speed of the driven shaft, rotation should be transmitted from a pulley of

larger diameter to one of a smaller diameter. On the contrary, if the speed of the driven shaft is to be decreased, its pulley should be of a larger diameter than that of the driving pulley.

The speed of the driven shaft n_{dn} is determined from the formula

$$n_{dn}=\frac{D_{dr}}{D_{dn}} \times n_{dr} \text{ rpm} \quad (8)$$

in which n_{dr} denotes the speed of the driving shaft in rpm.

D_{dn} and D_{dr} denote the diameters of the driven and driving pulleys, respectively, in mm.

The ratio $\frac{D_{dr}}{D_{dn}}$ is called the *transmission ratio* and is denoted by the letter i, therefore:

$$\frac{D_{dr}}{D_{dn}}=i \text{ and } n_{dn}=in_{dr} \text{ rpm.} \quad (9)$$

§ 15. CHAIN DRIVES

Roller chain (Fig. 31*a*) and silent chain (Fig. 31*b*) are employed in machine tool drives. The directions of the driving and driven sprockets coincide as in open belt drives. To determine the speed of the driven shaft n_{dn}, the speed of the driving shaft n_{dr} is multiplied by the ratio of the number of teeth on the driving sprocket Z_{dr} to the number of teeth on the driven sprocket Z_{dn}:

$$n_{dn}=\frac{Z_{dr}}{Z_{dn}} n_{dr} \text{ rpm.} \quad (10)$$

The ratio $\frac{Z_{dr}}{Z_{dn}}=i$ is called the *transmission ratio of the chain drive* and therefore:

$$n_{dn}=in_{dr} \text{ rpm.} \quad (11)$$

§ 16. TOOTHED GEARING

Toothed gearing (Fig. 32) serves for transmitting motion between *parallel*, *intersecting*, and *crossed* shafts. Spur, parallel helical or herringbone gears are used to transmit rotation between parallel shafts (Fig. 32 *a*, *b* and *c*). Straight or spiral bevel gears are used between intersecting shafts (Fig. 32 *e*). For crossed shafts, worm gearing is used (Fig. 32 *f*) or crossed helical gears.

Rotary motion is converted into reciprocating motion or vice versa by means of a pinion and rack (Fig. 32*g*) or by other mechanisms which will be described further on.

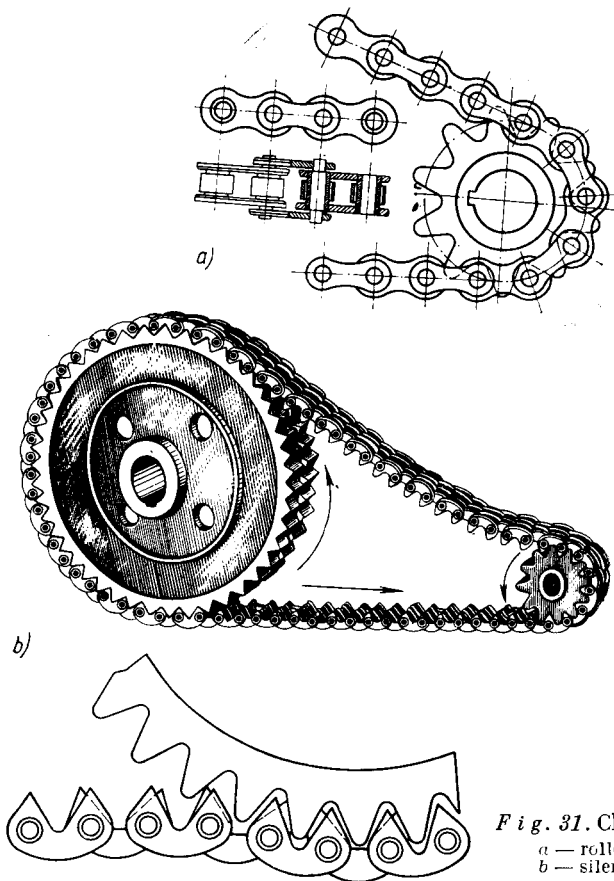



Fig. 31. Chain drives
a — roller chain,
b — silent chain.

The same formulas as for chain drives are used for determining the speed of a driven gear:

$$n_{dn}=\frac{Z_{dr}}{Z_{dn}}\ n_{dr}\ \text{rpm.}; \tag{12}$$

$$\frac{Z_{dr}}{Z_{dn}}=i;$$

$$n_{dn}=in_{dr}\ \text{rpm.} \tag{13}$$

The ratio $\frac{Z_{dr}}{Z_{dn}}=i$ is called the *gearing ratio.*

In external spur and helical gearing, the driving and driven shafts rotate in opposite directions (Fig. 33 *a*).

If two shafts are to rotate in the same direction, then an intermediate or idler gear (Fig. 33 *b*) is arranged between the gears on the

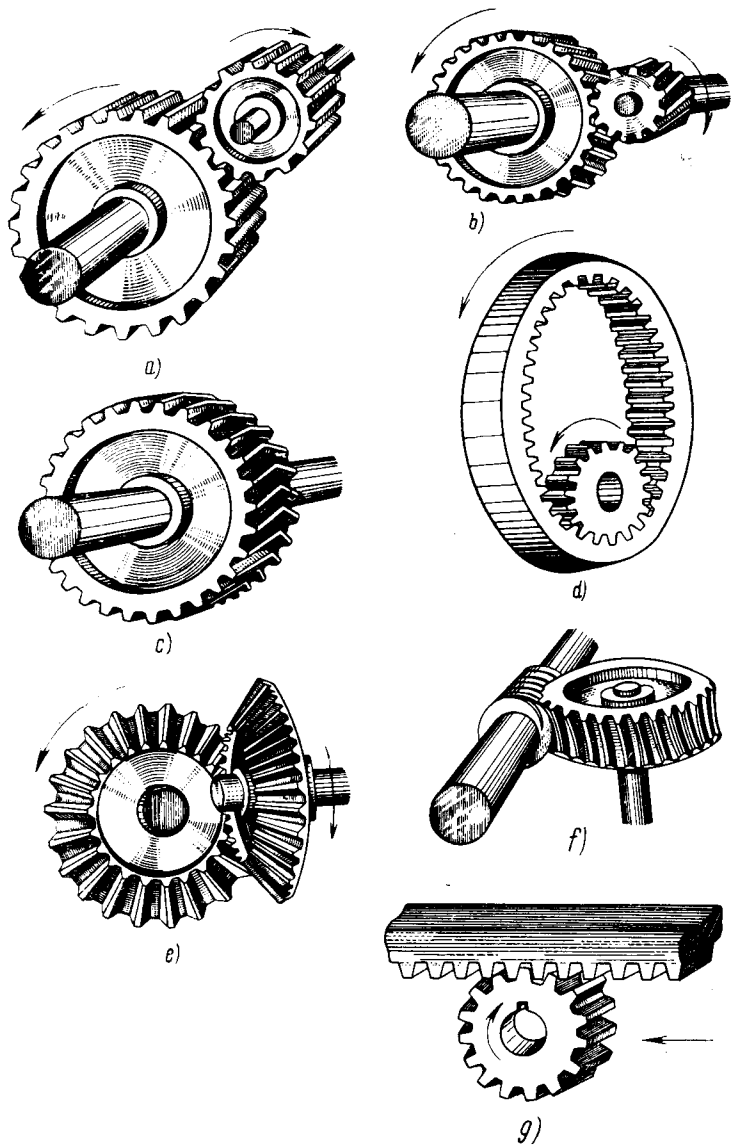



Fig. 32. Toothed gearing.

a — spur gears, *b* — parallel helical gears, *c* — herringbone gear, *d* — external and internal gears, *e* — straight bevel gears, *f* — worm and wheel, *g* — rack and pinion.

shafts or external and internal, spur or helical gears are used (Fig. 32 *d*).

The direction of rotation of bevel gears depends on their location. In Fig. 33 *c*, two bevel gears are mounted freely on the shaft. They mesh with a large bevel gear; the left-hand gear rotates in one direction and the right-hand gear, in the opposite direction. This feature is used to impart rotation alternately in either direction to a shaft by engaging the clutch to the corresponding gear.

Trains of gears are employed in gear boxes (Fig. 33 *d*). The speed of the last or fourth shaft (n_{IV}) of the gear box (Fig. 33 *d*) may be determined, after determining the speeds of all the intermediate shafts, according to formulas (12) and (13):

$$n_{II} = \frac{Z_1}{Z_2} n_I \text{ rpm}; \quad n_{II} = i_I n_I \text{ rpm},$$

in which

$$i_I = \frac{Z_1}{Z_2};$$

$$n_{III} = \frac{Z_3}{Z_4} n_{II} \text{ rpm}; \quad n_{III} = i_{II} n_{II} \text{ rpm},$$

in which

$$i_{II} = \frac{Z_3}{Z_4};$$

$$n_{IV} = \frac{Z_5}{Z_6} n_{III} \text{ rpm}; \quad n_{IV} = i_{III} \; n_{III} \text{ rpm},$$

in which

$$i_{III} = \frac{Z_5}{Z_6}.$$

To avoid such lengthy computations, the speeds of the intermediate shafts are introduced into the formula for calculating the speed of the last shaft:

$$n_{IV} = \frac{Z_1}{Z_2} \times \frac{Z_3}{Z_4} \times \frac{Z_5}{Z_6} \times n_I \text{ rpm}.$$

$$n_{IV} = i_I \times i_{II} \times i_{III} \times n_I \text{ rpm}.$$

Worm gearing is usually employed to sharply reduce the speed of the driven shaft on which the worm wheel is mounted since the gearing ratio i is very small. The direction of rotation of the worm wheel depends on the direction of rotation and the hand of the worm. The speed of the driven shatf is determined from the formula:

$$n_{dn} = \frac{a}{Z} \times n_{dr} \text{ rpm}, \tag{14}$$

$$\frac{a}{Z} = i \tag{15}$$

in which a denotes the number of threads or starts on the worm.
Z denotes the number of teeth on the worm wheel.

1. ASSEMBLING BELT DRIVE

Two main types of belt drive are used, ordinary (flat) and V-belt drives.

The ordinary belt drives may be with or without a tensioning device.

Ordinary belt drives without tensioning devices are grouped into open, crossed-belt and quarter-twist belt drives. The open drive operates with parallel shafts. This arrangement of shafts is the simplest and most convenient for belt driving and at the same time is best for the belt itself. A pulley is fitted to each shaft and the belt is slipped over the pulleys, both of which rotate in the same direction.

One of the two pulleys is the driving pulley (receiving motion from the shaft and transmitting it to the belt) and the other is driven (receiving motion from the belt and transmitting it to the shaft).

An open belt drive is illustrated in Fig. 169*a*. Pulley O_1 is the driving pulley as it receives rotary motion (clockwise as indicated by the arrow) from its shaft. The driven pulley O_2 will also rotate clockwise. Hence, in an open belt drive both the belts rotate in one direction.

If the direction of rotation required is opposite to that of the driving pulley, a crossed-belt drive is used (Fig. 169*b*). In the case illustrated the driving pulley O_1 turns clockwise and the driven pulley O_2 turns counterclockwise. The shafts are parallel to each other.

When the shafts are perpendicular to each other a quarter-twist belt drive is used (Fig. 169*c*).

The main disadvantage of belt drive is that the belt slips a certain amount, depending on the tension of the belt and the diameter of the arc with which the belt embraces the pulley. When the pulley diameters are different the small-

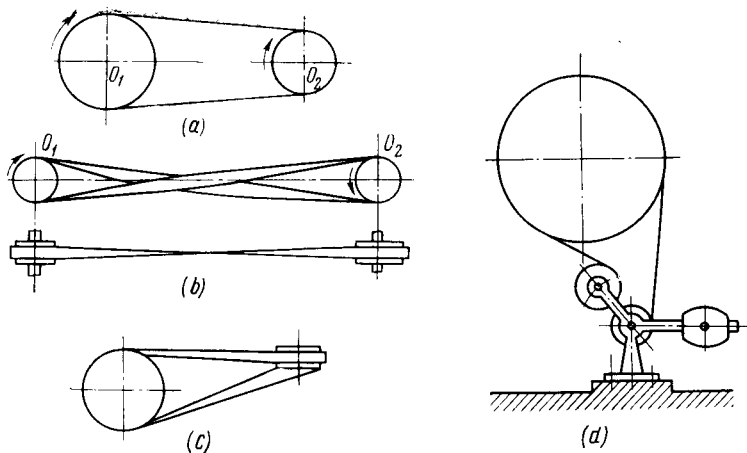



Fig. 169. Belt drive:
(*a*) open-type belt drive; (*b*) crossed-belt drive; (*c*) quarter-twist belt drive (*d*) position of tensioning rollers

er pulley is in a less advantageous position as it has a smaller contact area with the belt.

The tension and arc of pulley contact can be increased by using tensioning rollers (Fig. 169*d*).

Pulleys. Pulleys are usually cast from iron or steel. They are made either with arms or with a solid disk which can be perforated to reduce weight. Pulleys may be split or solid, the former being used when the pulley is arranged between bearings.

Single or stepped (with several steps of different diameter) pulleys are in use.

At present the V-belt drive is widely used. This type of drive has several advantages over flat drives:

(a) increase in the transmission ratio i (which is from 1/10 to 10 for V-belt drives);

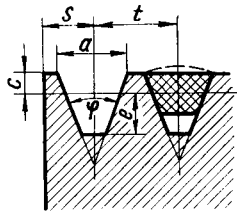
(b) decrease in the overall size of the drive (the permissible distance between the rims of the pulleys is 40-50 mm);

(c) increased friction coefficient due to the V-shape, and as a result, reduced pressure on the shafts;

(d) increased efficiency factor to 0.975-0.985.

Recommended dimensions for the V-belt pulley grooves are given in Table 19.

Table 19

Cross-Sectional Profile of V-belts

Elements of cross-section	Dimensions						
	1	2	3	4	5	6	7
a — groove width	10	13	17	22	32	38	50
e — groove height	10	13	17	22	30	36	48
c — distance from outer diameter to middle line	3	4	5	7	9	12	16
t — pitch (distance between grooves)	12	16	21	27	38	44	58
s — distance from pulley edge to 1st groove	9	12	15	18	23	26	32
φ — groove angle (from 34° to 40°)							

The relation between the rotational speeds of the driving and driven pulleys is determined by the transmission ratio which expresses the relation between the diameters of the driving and driven pulleys. The ratio shows whether the speeds are different and, if so, whether the speed of the driven pulley is greater or less than that of the driving pulley.

Let d_1 be the diameter of the driving pulley and d_2 that of the driven pulley, n_1 the rotational speed of the driving pulley and n_2 that of the driven pulley. The relation between these values can be expressed by the following equation:

$$\frac{n_2}{n_1} = \frac{d_1}{d_2}$$

Example. $d_1 = 450$ mm, $d_2 = 250$ mm, $n_1 = 200$ rpm

$$i = \frac{d_1}{d_2} = \frac{450}{250} = \frac{9}{5}; \qquad \frac{n_2}{n_1} = \frac{d_1}{d_2}; \quad n_2 = n_1 i = 200 \times \frac{9}{5} = 360 \text{ rpm}.$$

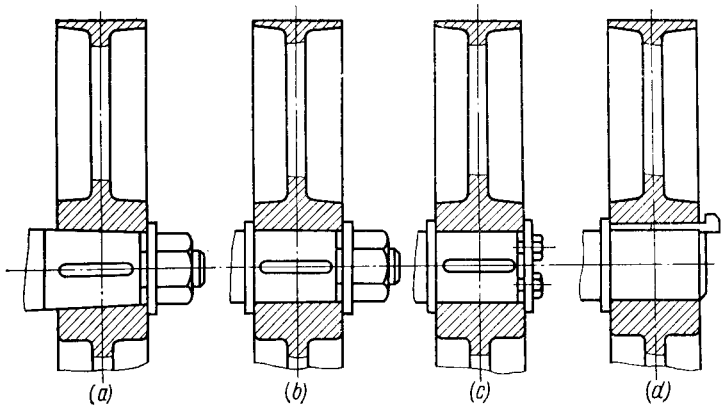



Fig. 170. Securing pulleys on end of shaft

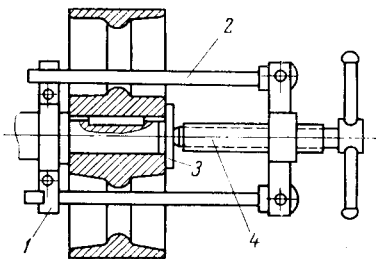



Fig. 171. Cramp

It will be noted that in determining the transmission ratio the relation between the driving pulley diameter and the driven pulley diameter is taken.

Assembling pulleys. Solid working pulleys are mounted on the shaft with tight or wringing fit. If the pulley is mounted on the end of the shaft, the end may be tapered (Fig. 170*a*), cylindrical (Fig. 170*b*, *c*, *d*) or fitted on feather or taper keys. The cylindrical shaft with a feather key is shouldered (Fig. 170*b*) to fix the pulley in position, and the pulley is additionally secured by a nut. No additional securing is required for the taper key fastening (Fig. 170*d*).

Before mounting the pulley on the shaft the key and keyways on the shaft and hub must be mutually fitted together, and the contact areas carefully cleaned and coated with machine oil.

Screw-type appliances such as cramps are recommended for fitting the pulley on the shaft. A detachable strap *1* (Fig. 171) is placed on the shaft resting against the shaft shoulder. Draw rods *2* are passed between the pulley arms. Plate *3* which has screw *4* pressing against it, is placed on the hub face of the pulley. By turning the screw and gently tapping the pulley at different points on the outside of the hub the pulley is gradually forced on the shaft. Tapping is intended to prevent the pulley seizing on the shaft due to distortions.

Split pulleys can be mounted at any point on the shaft. In assembling the pulley the two halves are joined and tightened by stud bolts.

The pulleys mounted on the shaft are checked for runout. Runout (or beat) is the fluctuation of the outer surface of a rotating part. The maximum fluctuation of the surface in one revolution determines the amount of the runout.

Runout of pulleys results in rapid wear of the bearings. On precise high-speed metal-working machines runout contributes to increased vibration which lowers the quality of the surface finish of the work.

The reasons for pulley runout are bent shafts, poor machining of the pulleys and distortions in fitting the pulley on the shaft.

The runout of pulleys is checked by a surface gauge with a scriber or by an indicator. If an indicator is used the reading is taken from the dial, and if a scriber tool is used the runout is measured with a clearance gauge.

The scriber should be fitted to a stationary stand (Fig. 172*a*) and the pulley turned so that the scriber just touches the pulley surface. In this position chalk is held against the scriber point and a line traced either on the pulley's end face or on the cylindrical portion (depending on which surface is being checked). This chalk mark will indicate the position of the pulley at maximum runout in the direction of the scriber. The pulley is then turned half a turn either way and the gap between the pulley surface and the scriber point is measured with a clearance gauge. This gap will indicate the maximum oscillation amplitude in turning the pulley, i.e., either the endwise δ_e or radial δ_r runout.

The positioning of the pulleys on the shafts is checked by the belt, which should not slip off the pulleys during operation.

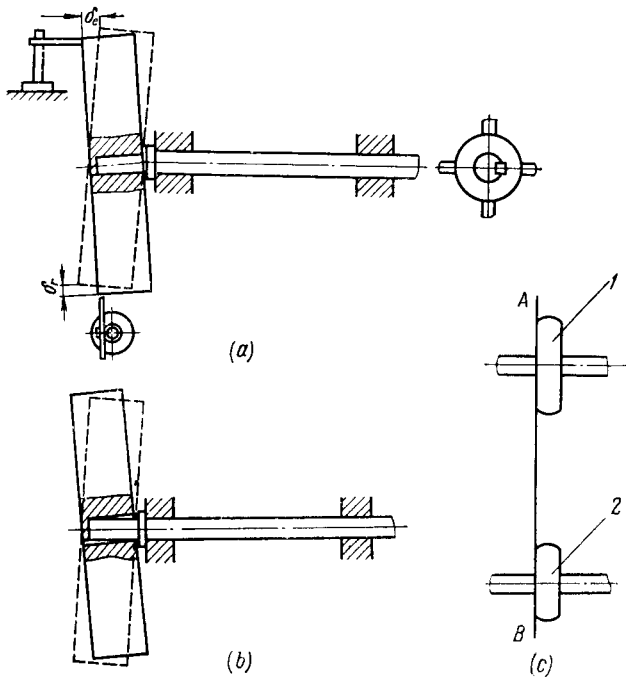



Fig. 172. Checking pulleys for end face and radial runout:
(*a*) shaft end bent; (*b*) key fitted wrongly; (*c*) checking the pulley for alignment with a stretched cord

Pulleys with equal rim widths should have their side surfaces in one plane. This can be checked with a stretched cord AB as shown in Fig. 172*c*. First stretch the cord and position it so that it just touches the edges of one pulley, say pulley *1*. Stretch the cord in a straight line towards pulley *2*. If the pulleys are correctly arranged the cord will touch both edges of pulley *2* and remain straight.

Types of Gear Drives. Spur gear drives are the simplest in design and manufacture. They transmit rotary motion between parallel shafts.

The gear wheel fitted on the driving shaft is called the driving gear wheel and the wheel it meshes with is called the driven gear wheel.

The transmission gear ratio i indicates the relation between the number of teeth of the driving wheel z_1 and the number of teeth of the driven wheel z_2, or between the rotational speeds of the driven wheel n_2 and the driving wheel n_1, i.e.,

$$i = \frac{n_2}{n_1} = \frac{z_1}{z_2}$$

Example. A driving wheel has $z_1 = 25$ teeth and rotates at $n_1 = 80$ rpm. The driven wheel has $z_2 = 80$ teeth. The rotational speed of the driven wheel is required.

The transmission ratio

$$i = \frac{25}{80} = \frac{5}{16}$$

The driven wheel rotates at

$$n_2 = z_2 i = 80 \times \frac{5}{16} = 25 \text{ rpm}$$

The transmission ratio of a worm gear

$$i = \frac{z_s}{z_t}$$

in which z_s = number of starts on worm

z_t = number of teeth on wheel.

The transmission ratio of a worm gear is the ratio of starts on the worm to the number of teeth on the wheel.

Depending on the number of conjugated pairs of gear wheels gear drives are grouped into one-, two-, three-, etc., stage drives.

59. Classification of Transmission Mechanisms

The transmission of energy from one machine to another or from one member to another in a machine is accomplished by means of various transmission mechanisms.

The types of drives may be classified according to various characteristics.

Classification of drives according to *the means for transmitting motion.*

1. Drives in which motion is effected by using frictional forces (friction drive, belt drive, rope drive).

2. Drives in which motion is accomplished by direct geometric engagement of one member of a kinematic pair with the other (gear drive, screw drive, worm drive, chain drive).

Classification of drives according to *the nature of transmission members between the driving and driven members.*

1. Motion is transmitted through direct contact between the driving and driven members (friction drive, gear drive, screw drive, worm drive).

2. Motion is transmitted through an intermediate link connecting the driving and driven members (belt drive, rope drive, chain drive, band transmission).

60. Gear Ratio

A shaft and transmission members (wheels or pulleys) mounted on it and transmitting a given torque are referred to as *driving*, while members receiving the torque and the motion from the driving members are termed *driven.*

Symbols pertaining to driving members of a direct drive (Fig. 108*a*) and a continuous drive (Fig. 108*b*) are assigned the subscript *1*, and for driven members, the subscript *2*.

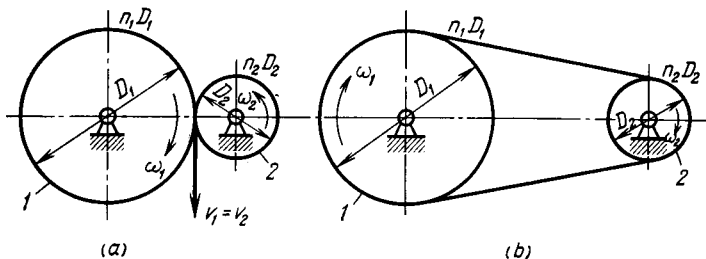



Fig. 108.

The peripheral speed of a driving member can be found from the formula

$$v_1 = \frac{\pi D_1 n_1}{60} \text{ m/sec.}$$

The peripheral speed of a driven member is

$$v_2 = \frac{\pi D_2 n_2}{60} \text{ m/sec.}$$

For transmission without slipping the peripheral speeds of the two members must be equal

$$v_1 = v_2$$

or

$$\frac{\pi D_1 n_1}{60} = \frac{\pi D_2 n_2}{60}.$$

Cancelling, we obtain

$$D_1 n_1 = D_2 n_2,$$

whence

$$\frac{n_1}{n_2} = \frac{D_2}{D_1}.$$

Here D_1 and D_2 are the diameters of the driving and driven members, n_1 and n_2 their numbers of revolutions, and v_1 and v_2 their peripheral speeds.

Since the ratio of the number of revolutions of the driving member to that of the driven member can be replaced by the

ratio of their angular velocities, i.e.,

$$\frac{n_1}{n_2}=\frac{\omega_1}{\omega_2},$$

we have

$$\frac{\omega_1}{\omega_2}=\frac{n_1}{n_2}=\frac{D_2}{D_1}=\frac{R_2}{R_1}.$$

The ratio of the number of revolutions or the angular velocity of the driving shaft to the number of revolutions or the angular velocity of the driven shaft is called the *gear* (or *transmission*) *ratio* and is denoted by i. For the case under consideration

$$i=\frac{\omega_1}{\omega_2}=\frac{n_1}{n_2}=\frac{D_2}{D_1}, \tag{120}$$

where ω_1 = angular velocity of driving disk,
ω_2 = angular velocity of driven disk.

The gear ratio is assigned a sign according to the sense of the angular velocities. Thus, in the mechanism represented in Fig. 108*a* the angular velocities are opposite and so have different sign. Therefore the gear ratio i_{1-2} has a minus sign, i.e.,

$$i_{1-2}=-\frac{\omega_1}{\omega_2}=-\frac{n_1}{n_2}. \tag{121a}$$

For the mechanism shown in Fig. 108*b* the angular velocities ω_1 and ω_2 have the same sense and therefore the ratio i_{1-2} is assigned a plus sign

$$i_{1-2}=\frac{\omega_1}{\omega_2}=\frac{n_1}{n_2}. \tag{121b}$$

The sign rule adopted above for the gear ratio is conventional.

64. Belt Drives: Fundamental Concepts

If the driving shaft is at a considerable distance (up to 15 m) from the driven shaft, the moment of forces can be transmitted by a flexible connector using the frictional forces developed between the surface of the pulley and the flexible body.

Fig. 112.

Belts and ropes are used as a flexible connector for this kind of drives (Fig. 112*a* through *d*).

The peripheral speed on the driving pulley (Fig. 112*a*) is

$$v_1 = \frac{\pi D_1 n_1}{60}.$$

The peripheral speed on the driven pulley is

$$v_2 = \frac{\pi D_2 n_2}{60}.$$

If no account is taken of slipping between the belt and the pulley, the speeds v_1 and v_2 must be equal

$$v_1 = v_2$$

and

$$\frac{\pi D_1 n_1}{60} = \frac{\pi D_2 n_2}{60}.$$

Cancelling, we obtain

$$D_1 n_1 = D_2 n_2 \quad \text{or} \quad \frac{n_1}{n_2} = \frac{D_2}{D_1}.$$

In practice $v_1 \neq v_2$ since some elastic slippage always occurs between the belt and the pulley rim. Usually 1 to 3 per cent of the peripheral speed is lost in slipping. Introducing this correction, we obtain

$$v_2 = (1 - \varepsilon) v_1,$$

where ε is the slip ratio ($\varepsilon = 0.01$ to 0.03). Then

$$i_{1-2} = \frac{n_1 (1 - \varepsilon)}{n_2} = \frac{D_2}{D_1}. \qquad (127)$$

Elastic slippage is unavoidable in the operation of a belt drive under loading. This is not to be confused with parasitic slippage due to overloading or poor design.

65. Gear Drives: General Considerations

A gear drive differs from a friction drive in that both wheels (rollers) are provided with teeth of definite shape. When one of the wheels rotates its teeth engage the teeth of the mating wheel, setting it in rotation. A gear drive has a constant gear ratio.

Gear drives are the most common mechanisms for transmission of rotary motion and are capable of transmitting considerable power.

Gear drives are simple in design and highly efficient.

The following types of gear drives are distinguished according to the relative position of the geometric axes of the driving and driven shafts:

(a) spur gearing employed when the shaft axes are parallel,
(b) bevel gearing employed when the shaft axes intersect,
(c) helical gearing and worm gearing employed when the shaft axes cross in space.

66. Gear Ratio of a Spur Gear Drive

Figure 113 shows a sketch of spur gearing. Let ω_1 denote the angular velocity of the driving wheel and ω_2 the angular velocity of the driven wheel

$$\omega_1 = \frac{\pi n_1}{30}, \quad \omega_2 = \frac{\pi n_2}{30}.$$

The *gear ratio* is the angular velocity or the number of revolutions of the driving wheel divided by the angular

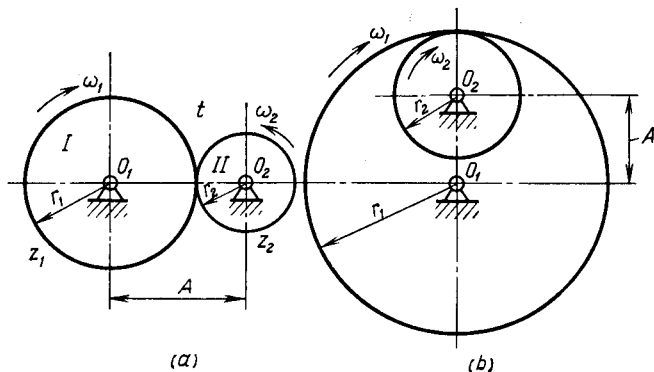



Fig. 113.

velocity or the number of revolutions of the driven wheel

$$i_{1-2} = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2}.$$

But the numbers of revolutions are inversely proportional to the radii of the wheels, therefore

$$i_{1-2} = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2} = \frac{r_2}{r_1}.$$

Let z_1 denote the number of teeth of the driving wheel, z_2 the number of teeth of the driven wheel, t the tooth pitch.

The length of the circumference of the driving wheel is $2\pi r_1 = z_1 t$.

The length of the circumference of the driven wheel is $2\pi r_2 = z_2 t$.

Taking the ratio of the length of the circumference of the driven wheel to the length of the circumference of the driving wheel, we obtain

$$\frac{2\pi r_2}{2\pi r_1} = \frac{z_2 t}{z_1 t}$$

or

$$\frac{r_2}{r_1} = \frac{z_2}{z_1}.$$

The gear ratio for spur gears is then

$$i_{1-2} = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2} = \frac{r_2}{r_1} = \frac{z_2}{z_1}. \qquad (128)$$

Consequently, the gear ratio for a gear drive can also be expressed as the number of teeth of the driven wheel divided by the number of teeth of the driving wheel.

Besides external gearing (Fig. 113*a*), mechanical engineering employs internal gearing (Fig. 113*b*). In contrast to a pair of external wheels, internal wheels rotate in the same direction.

67. Gear Trains

Two cases are distinguished in series-connected gearing:
(a) a train with idler wheels—in-line connection;
(b) a train with multiple engagement—multistep, or compound, gearing.

In-Line Connection. If the driving shaft is at a considerable distance from the driven shaft, intermediate gears have to be inserted which do not affect the overall gear ratio of the drive (Fig. 114).

For instance, we have the gear ratio

$$i_{1-4} = \frac{\omega_1}{\omega_4} = \frac{r_4}{r_1}.$$

A wheel of radius r_1 is fitted on the driving shaft, and a wheel of radius r_4 on the driven shaft.

If an arbitrary number of gears are mounted between the driving and driven wheels, then in order to preserve the gear ratio the intermediate gears (called idler gears) should have the same tooth pitch as the driving and driven wheels.

We determine the overall gear ratio of the drive.

The gear ratio of the first pair of engaging gears is

$$i_{1-2}=\frac{\omega_1}{\omega_2}=\frac{r_2}{r_1}.$$

For the second pair

$$i_{2-3}=\frac{\omega_2}{\omega_3}=\frac{r_3}{r_2}.$$

For the third pair

$$i_{3-4}=\frac{\omega_3}{\omega_4}=\frac{r_4}{r_3}.$$

By multiplying the several gear ratios, we find

$$i_{1-2}i_{2-3}i_{3-4}=\frac{\omega_1}{\omega_2}\frac{\omega_2}{\omega_3}\frac{\omega_3}{\omega_4}=$$

$$=\frac{r_2}{r_1}\frac{r_3}{r_2}\frac{r_4}{r_3}.$$

After cancelling we obtain

$$i_{1-2}i_{2-3}i_{3-4}=\frac{\omega_1}{\omega_4}=\frac{r_4}{r_1}.$$

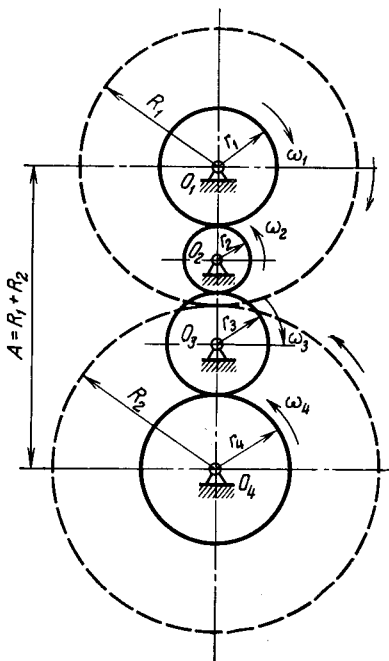
Fig. 114.

The result obtained is the assigned overall gear ratio of the drive, consequently,

$$i_{1-2}i_{2-3}i_{3-4}=\frac{\omega_1}{\omega_4}=\frac{r_4}{r_1}=i_{1-4}. \tag{129}$$

We arrive at the conclusions:

1. *The overall gear ratio of an in-line connection of engaging gears is equal to the product of the individual gear ratios of the drives forming the series.*

2. *The overall gear ratio is equal to the angular velocity of the first wheel divided by the angular velocity of the last wheel or to the inverse ratio of their radii.*

3. *Intermediate (idler) wheels do not change the overall gear ratio, but they affect the direction of rotation of the driven shaft: (a) with an even number of idler wheels the driving and driven shafts rotate in opposite directions; (b) with an odd number of idler wheels the direction of rotation is the same.*

The incorporation of a set of small gears in place of the two gears shown dashed in Fig. 114 does not change the assigned

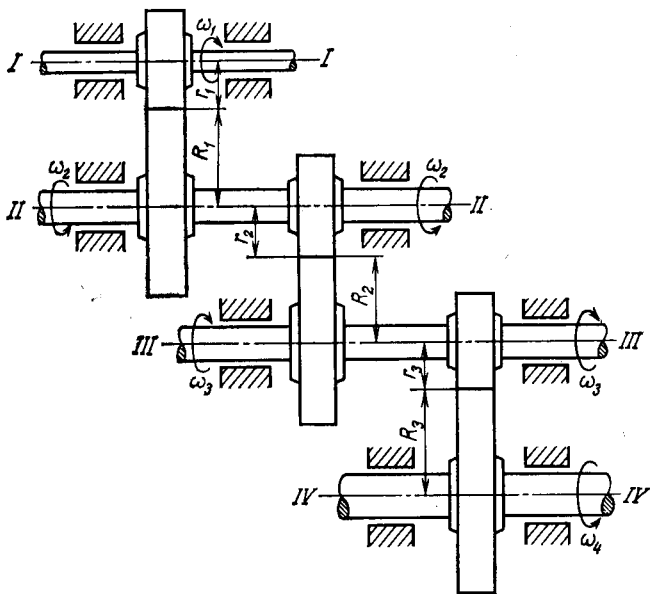



Fig. 115.

gear ratio, facilitates transmission, reduces the dimensions (this being very important in machines) and makes it possible to change the direction of rotation of the driven wheel.

Multistep Gearing. A large gear ratio cannot be obtained with just one pair of gears. To increase it, use is made of a set of series-connected gears called a multistep gear (Fig. 115). A multistep gear is characterized by the fact that two gears are mounted on each intermediate shaft.

The first wheel of the drive is fitted on the driving shaft, and the last on the driven one.

The gear ratio of the first engaging pair of gears is

$$i_{1-2}=\frac{\omega_1}{\omega_2}=\frac{R_1}{r_1}=\frac{Z_1}{z_1}.$$

For the second pair

$$i_{2-3}=\frac{\omega_2}{\omega_3}=\frac{R_2}{r_2}=\frac{Z_2}{z_2}.$$

For the last pair

$$i_{3-4}=\frac{\omega_3}{\omega_4}=\frac{R_3}{r_3}=\frac{Z_3}{z_3}.$$

By multiplying the individual gear ratios, we find

$$i_{1-2}i_{2-3}i_{3-4}=\frac{\omega_1}{\omega_2}\,\frac{\omega_2}{\omega_3}\,\frac{\omega_3}{\omega_4}=\frac{R_1}{r_1}\,\frac{R_2}{r_2}\,\frac{R_3}{r_3}=\frac{Z_1}{z_1}\,\frac{Z_2}{z_2}\,\frac{Z_3}{z_3}$$

or

$$i_{1-2}i_{2-3}i_{3-4}=\frac{\omega_1}{\omega_4}=\frac{R_1R_2R_3}{r_1r_2r_3}=\frac{Z_1Z_2Z_3}{z_1z_2z_3}.$$

The ratio of the angular velocity of the first driving wheel to the angular velocity of the last driven wheel is the gear ratio of the whole set of gears, i.e.,

$$i_{1-2}i_{2-3}i_{3-4}=\frac{\omega_1}{\omega_4}=i_{1-4}=\frac{R_1R_2R_3}{r_1r_2r_3}=\frac{Z_1Z_2Z_3}{z_1z_2z_3}. \qquad (130)$$

Consequently, the gear ratio of the whole series-connected set of gears can be found by dividing the product of the radii of the driven wheels by the product of the radii of the driving wheels.

This arrangement of gears makes it possible to greatly increase (or reduce—less frequently) the overall gear ratio of the drive.

In the particular case when the radii of the driving wheels are equal and so are the radii of the driven wheels, i.e.,

$$r_1=r_2=r_3=r,$$

$$R_1=R_2=R_3=R,$$

the gear ratio of the drive is

$$i=\left(\frac{R}{r}\right)^3 \text{ or } i=\left(\frac{Z}{z}\right)^3.$$

In the general case this formula becomes

$$i = \left(\frac{R}{r}\right)^k = \left(\frac{Z}{z}\right)^k,$$

where k = number of pairs of gears forming the drive,
Z = number of teeth of driven wheel,
z = number of teeth of driving wheel.

68. Worm Gearing

General Considerations. A worm and worm wheel are used to transmit motion with a constant gear ratio between shafts crossing in space. We shall consider the most common worm drives with shafts crossing at right angles.

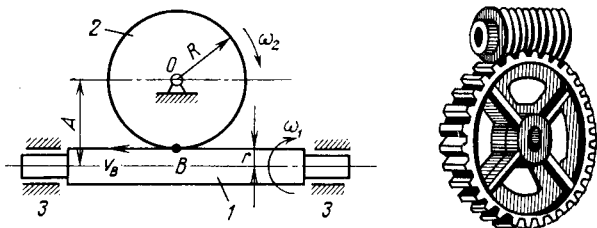



Fig. 116.

A worm drive (Fig. 116) consists of a worm (screw) *1* fitted on a shaft or made integral with the shaft, a worm wheel *2* mounted on another shaft, and bearings *3* supporting the shafts of the worm and worm gear.

Let us first see how a screw is generated.

The unwrapped circumference of the base of a right circular cylinder represents a straight line $AB = 2\pi r$ (Fig. 117*a*). Drawing from point A a sloping straight line at an angle λ to the base, we cut off a segment $BC = h$ on the side of the rectangle. Wrapping the sloping straight line AC around the cylinder, we obtain a continuous helical line.

The height h is called the *lead of the helix*, i.e., the axial distance that a point on the helix will move in one revolution, and the angle λ is termed the *lead angle*. From Fig. 117*a*

it is seen that

$$h = 2\pi r \tan \lambda,$$

$$\tan \lambda = \frac{h}{2\pi r}.$$

Let us take a triangle the plane of which contains the axis of the cylinder and the base coincides with one of the generators. Displace the triangle so that its vertex moves along a helical line on the lateral surface of the cylinder. Then the sides of the triangle which include the vertex will describe helical surfaces which form a screw with a triangular thread.

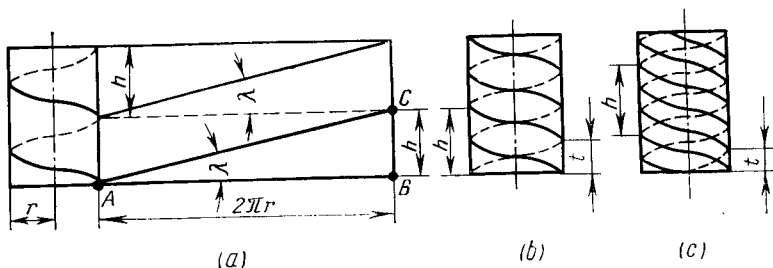



Fig. 117.

If the same motion is imparted to a rectangle, a screw with a square thread is formed. If there are two or three threads along one lead of a helix, the screw is referred to as double-threaded (Fig. 117*b*) or triple-threaded (Fig. 117*c*), respectively.

A body surrounding a screw the top of whose thread coincides with the thread groove of the screw is called a nut.

Gear Ratio. The gear ratio for a worm and worm wheel drive is defined by the same formula as for gear drives

$$i_{1-2} = \frac{z_2}{z_1},$$

where z_2 is the number of teeth in the wheel, z_1 the number of threads (or teeth) wrapped on the worm (in practice $z_1 \leqslant 5$ since when the number of threads is increased the lead angle becomes larger and this intensifies the wear of the worm).

In terms of the geometric elements of a worm and worm wheel the gear ratio is determined as follows. The linear

velocity of a point of the thread on the base cylinder along the axis of the worm is

$$v_1 = \frac{hn_1}{60},$$

where h is the axial distance that a point on the helix of the worm will move in one revolution of the worm, n_1 the number of revolutions of the worm per minute.

The velocity of the teeth on the base circle of the worm gear is

$$v_2 = \frac{\pi D n_2}{60},$$

where D is the diameter of the worm wheel, n_2 the number of revolutions of the wheel per minute.

Since the velocities must be equal, we have

$$\frac{hn_1}{60} = \frac{\pi D n_2}{60}$$

or

$$\frac{n_1}{n_2} = \frac{\pi D}{h} = i_{1-2}.$$

It is known that

$$h = 2\pi r \tan\lambda = \pi d_1 \tan\lambda,$$

consequently,

$$i_{1-2} = \frac{n_1}{n_2} = \frac{\omega_1}{\omega_2} = \frac{\pi D}{h} = \frac{\pi 2R}{2\pi r_1 \tan\lambda} = \frac{R}{r_1 \tan\lambda},$$

where r_1 and d_1 are the radius and diameter of the worm, respectively.

It is also known that

$$\pi D = z_2 t,$$

where z_2 is the number of teeth in the worm wheel, t the pitch.

$$h = z_1 t,$$

where z_1 is the number of threads on the worm.

In the general form the gear ratio is

$$i = \frac{n_1}{n_2} = \frac{\omega_1}{\omega_2} = \frac{\pi D}{h} = \frac{R}{r_1 \tan\lambda} = \frac{z_2 t}{z_1 t} = \frac{z_2}{z_1}. \qquad (131)$$

Consider two examples of the kinematic analysis of transmission mechanisms.

1. *Spur Gear*. The gear ratio is $i = 2$. The number of revolutions of the driving shaft is $n_1 = 800$ rpm, the number of teeth of the driving wheel $z_1 = 30$.

We determine the number of revolutions and the number of teeth of the driven wheel

$$n_2 = \frac{n_1}{i} = \frac{800}{2} = 400 \text{ rpm},$$

$$z_2 = z_1 i = 30 \times 2 = 60.$$

2. *Worm and Worm Wheel*. The worm is double-threaded, i.e., $z_1 = 2$, the number of revolutions of the worm shaft is $n_1 = 2{,}000$ rpm, the gear ratio $i = 40$.

We determine the number of teeth of the worm wheel

$$i = \frac{z_2}{z_1}, \quad z_2 = iz_1 = 40 \times 2 = 80.$$

The number of revolutions of the driven shaft per minute is

$$n_2 = \frac{n_1}{i} = \frac{2{,}000}{40} = 50 \text{ rpm}.$$

BELTING. Belts for power transmission may be made either from leather, rubber, canvas, or thin sheet steel. Leather belts are, by far, the most commonly used. Rubber belts are used and the belt is exposed to the weather conditions or to the action of steam, because they do not stretch as easily as leather belts, under these conditions. Canvas belting is used when the materials in contact with the belt and the surrounding atmosphere would affect a leather or rubber belt. Steel belts made from thin flat sheets have been introduced within comparatively recent years. The advantages claimed are the greater amount of power transmitted for the same width of belt, the fact that steel belts are not influenced by the humidity of the air, and the low initial cost which is claimed to be less than that of leather or rubber belts of equal power transmitting capacity.

Application of Belts. — Whenever practicable, belts should be installed so that the slack side is above and the driving side below the pulleys. If this condition is reversed and the slack side is below, the arc of contact is materially lessened. In the case of leather belts, these should be placed on the pulleys with the hair or grain side next to the pulley rims. When installing all classes of belts, it is also important to provide the right amount of tension. If the belts are too tight, there is an unnecessary loss of power from excessive friction in the bearings and the belt is over-strained. On the other hand, if there is insufficient tension, the belt tends to flop and may run from the pulleys, especially if the load is suddenly applied. Ordinarily, the belt should have a slight amount of slackness on one side and the slacker the belt is up to a certain point, the greater the economy. It must be remembered, however, that certain kinds of belting are affected by the weather conditions, and lengthen or shorten according to the amount of moisture in the atmosphere.

Two shafts connected by a belt should never be placed one directly above the other, if this can be avoided, because the belt usually must be kept very tight in order to transmit a normal amount of power. Belting is frequently ruined because shafts or pulleys are not in alignment. Frequently the belts are kept on the pulleys either by means of guides or they rub against the hangers or parts of the machine until the belt is injured. Sometimes the shafting is thrown out of alignment by a rearrangement of machines, causing variations in the floor load. It is advisable to inspect the shafting for alignment occasionally as numerous tests have demonstrated that there is a considerable loss of friction in many shops and factories where the importance of shaft alignment is not appreciated. A mistake is sometimes made of locating shafts too close together. There should be sufficient distance between them to allow the belt to recover somewhat from the strain applied on the tight or working side. While this distance depends upon the size of the belts and pulleys, it should be sufficient to allow a slight sag to the slack side of the belt. On the other hand, if the shafts are too far apart, the weight of the belt increases the load on the bearings and the initial cost of the belting is unnecessarily increased. Excessive center-to-center distances also cause the belt to sway laterally, which is injurious to the belt. Pulleys of fairly large diameter should be used, if possible, as this means economy in the transmission of power, and less wear and tear on the belting.

Belts and pulleys should be kept clean and free from accumulations of dust and grease, and particularly lubricating oils, some of which permanently injure the leather. They should be well protected against water, and even moisture, unless especially waterproofed. Resin should not be used to prevent belts from slipping. If a belt slips, examine the pulley for dirt. Clean all the dirt from both the pulley and belt, and rub the pulley surface of the belt with a dressing composed of 2 parts of tallow and 1 part of fish oil, rendered and allowed to cool before using. This will soften a belt and also preserve it, and it will not build up on the pulley and cause the belt to run to one side. If the belt then slips, it is due to overloading, and the remedy lies in a leather-covered pulley, a wider belt, or a larger pulley.

The ability of a belt to transmit power depends upon the tension under which it is run, the degree of friction between the belt and the pulley, the complete contact of the belt

with the pulley, the speed of the belt, and the arc of the pulley in contact with the belt. The tensile strength of single, ordinary tanned leather belting is about 4000 pounds per square inch. According to some authorities, the working strain should not exceed 10 per cent of the tensile strength of the belt. The average leather belt will not transmit a force equal to its strength, for the reason that it will slip on its pulley before it will break.

Selection of Belt Stock. — In the manufacture of the best grades of leather belting, a comparatively small section of a hide is used, this being a strip about 30 inches wide extending from the tail to a point just back of the shoulder, its length being about four feet. (See Fig. 1.) On account of the difficulty of telling the grade of a belt except by use, certain leather belt makers are not over scrupulous as to the grades of leather entering their product, so that not only is the good section used, but flanks and other undesirable parts which are doubled with first grades in double belting. A belt made up in this manner may not be inferior in appearance to a belt made only from first-class parts, but, in use, it is bound to give inferior service. The use of the poorer grades with the best grades of leather belting is particularly bad. The inferior grades soon stretch, throwing almost the entire stress of belt pull on the superior grade. This uneven tension quickly deteriorates the belt. Probably a belt made up in this manner is inferior to that made of the poorer grades throughout. Making the belt of inferior grades throughout has the merit of equalizing the stretch, keeping both parts in even tension.

Oak-tanned leather is often considered the best for belting, although many high-grade belts are no longer tanned by the use of oak bark. Assuming that a good grade of leather is used, uniformity in the material is of first importance; that is, the different sections of which the belt is made should all be of the same grade. The belts should also be thoroughly stretched so that they do not have to be "taken up" every few days. That part of the hide extending along the spine and for some distance down the sides is firm and close in texture and the strongest for a belt. If the leather is taken too far down the side, it will be flexible and lack strength and closeness of texture. If the strips are cut too long, the ends will be taken from the neck of the animal, which is also inferior stock. A "short lap" belt is one made entirely from that part of the hide which comes from the back of the animal and the strips are not long enough to include any portion of the neck stock.

Manufacture of Leather Belting. — The quality of belting depends to a very great degree upon the methods of its manufacture. The treatment of the hide from which a belt is made, from the very beginning, has a marked influence upon the strength and wearing qualities of the belt. The principles of the manufacture of belting were reviewed in a paper read by Mr. F. H. Small before the American Society of Mechanical Engineers; a brief abstract of this paper is given in the following paragraphs. It is important that the user of belts should know something of the advantages secured by different methods of belt manufacture.

Preliminary Treatment. — Two methods of preserving or curing "green" hides are commonly used; *viz.*, drying and salting. The first is ordinarily too uncertain to permit of the use of hides cured in this way for belting, and practically all hides tanned for belting are what are known as "green-salted" hides. If the belt manufacturer tans the leather which he makes into belting — and this is the ideal arrangement — he buys as far as possible green-salted hides, free from scratches, brands, and grubs, short-haired, of as uniform thickness as obtainable, and which have been skillfully taken off, so that they are free from butcher cuts; *i.e.*, the manufacture of first quality belting begins with the purchase of first quality hides. Before being used for belting, these hides must be tanned, *i.e.*, so treated that they will not decompose or spoil and will remain flexible. There are various methods of tanning hides, but whatever the method used, the first steps, usually spoken of as the "beam house" treatment, are essentially the same; namely, preliminary washing with water to remove dirt and the salt or other material which may have been used to cure the hide; a cleaning of the flesh side to remove superfluous flesh, fat, etc., left on in the flaying — a soaking in milk of lime, or some other depilatory solution, to loosen

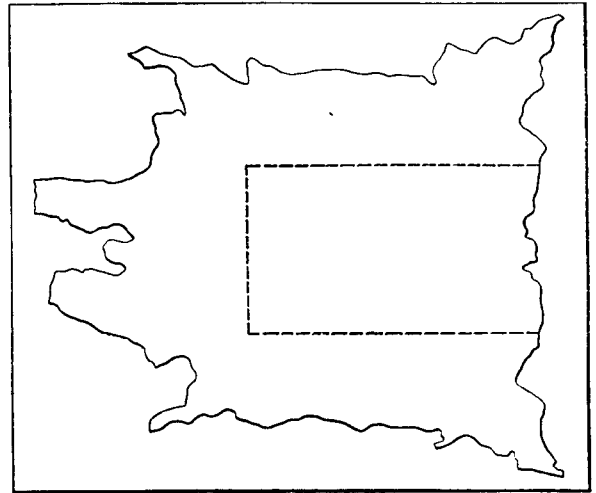



Fig. 1. Diagram showing Portion of Hide used in Making High-grade Leather Pelting

the hair, which is then pushed off by machine or with a dull two-handled knife; and a final washing to remove at least in part the unhairing chemical and to clean the hide. The quality of the finished leather depends, in a very large measure, upon the successful performance of these apparently simple operations.

Tanning. — The conversion of the hide into leather may be brought about through the use of any one of a variety of tanning materials, the peculiar characteristics of the finished leather being governed by the material used. One of the simplest and oldest methods of tanning is to cover the raw hide in the moist condition with grease and then continually manipulate it as it dries, thus working the grease into the fibers of the hide and producing a greased-tanned leather. Grease-tanned leather has little more fullness or body than the original hide, but it is exceptionally strong. Lace used for lacing belting together is tanned in this way. Another old method of tanning is to work into the hide a mixture of alum and salt. This gives a somewhat fuller leather than grease and is used before the grease treatment by some tanners of belt lace, because of the fullness gained, but it is at the expense of the toughness and wearing qualities of the lace.

Chrome-tanned Leather. — A modern tannage of somewhat similar nature is that with salts of chromium, the hide being immersed in a solution of one of these salts. Chrome-

tanned leather is fuller than alum-tanned, and much more permanent. Water will seriously injure alum-tanned leather, causing it to revert nearly to the condition of the raw hide and to become hard and cracky, whereas water has little or no effect on chrome leather. Chrome leather may even be immersed in boiling water for a short time without serious damage to the leather. Most of the "steam-proof" belting is made from chrome-tanned leather.

Vegetable Tannin. — By far the greatest percentage of heavy leather, and, in particular, that used for belting, is tanned with a tannin derived from some vegetable material. Years ago, practically all vegetable-tanned heavy leather in the United States was made with either hemlock or oak bark, or a mixture of the two. Now, while the old names, "hemlock," "oak," and "union," are still retained, it would be difficult to find a tanner making leather with these materials alone. The present-day tanner uses not merely hemlock and oak, but chestnut wood, or mimosa, or wattle, etc., some twenty-five or thirty materials being commercially available. The objection to practically all the tannages, aside from that with vegetable tannin, is that they do not make a plump, full, solid leather. The leather produced by them is tough, but thin and open. The original hide constitutes much the largest percentage of the finished product, only a small quantity of the tanning material remaining in the leather. This is in marked contrast to the vegetable tannage, in which so much tannin and other matter is deposited in and on the fibers of the hide that the original hide constitutes less than 50 per cent of the final leather.

Comparison of Results obtained by Different Methods. — Lack of firmness is a serious deficiency in a leather to be used for belting. Nevertheless, leathers made with some of the non-vegetable materials have sufficiently valuable properties, so that they have made considerable headway for belting even despite their failings. Chrome leather, for instance, can be produced in a comparatively short time. It will run practically unharmed in a temperature where vegetable-tanned leather would revert to a brittle, formless mass. It can be made exceptionally flexible, and has a high coefficient of friction. Consequently, belting from chrome leather has found a place for itself, which would be larger were it not for the deficiencies resultant from the lightness of the tannage, the lack of solidity of the leather, and the necessarily high cost, because of the small leather-yield. A leather tanned with a combination of alum and gambier, the gambier being used to supply the deficiencies of the alum as a filling material, has also been used. It is tough and pliable, and has given good service for high-speed work.

Preparing the Leather for Belting. — Of the hides tanned, only 50 per cent of the surface of the hide may legitimately be cut up into belting, and less than 40 per cent will go into first quality belting. The part suitable for belting is first curried, *i.e.*, given a supplementary grease-tannage; "set out" — to give a smooth flat piece of leather — and then stretched. The stretching is done on frames in which the wet leather may be clamped and subjected to as much tension as desired, the leather being allowed to dry under tension on the frames, so that an additional stretch resultant upon the natural shrinkage of the leather in drying is imparted to the leather. Before being stretched, the piece is usually cut into a center and two side pieces, inasmuch as the center portion, being more close fibered, will not stretch as much as the side pieces, and, by being divided, the leather can be stretched more in accordance with its capacity. After stretching, the leather is rolled and glossed to improve its looks, and is then ready to go to the stock-room.

Seasoning. — When the leather is received in the stock-room, it is sorted according to thickness into extra heavy, heavy, medium, and light, and then packed for future use. The sides are packed in square piles, alternate layers at right angles, to keep the stock flat and straight and allow of a circulation of air through the pile, thus aiding and hastening the seasoning process. This seasoning is an often neglected, but most desirable, operation, for the use of well-seasoned leather is as important in the manufacture of belting as the use of well-seasoned lumber is in building. Belts made from well-seasoned stock stretch less and more uniformly, retain their elasticity, and wear longer than belts from green stock.

The Belt Shop. — From the stock-room the leather goes to the belt shop, where its manufacture into belting takes place. The first step in this process is to straighten one edge of the leather. Next it is cut into strips of various widths by passing between a rapidly revolving circular knife and a guide, the strips being graded for width, and roughly for quality, as they come from the knife, and are then stored in racks. From these racks the leather goes to the sorters, by whom it is most carefully graded, both for thickness and quality, and on their expertness depends the maintenance of the standard set for each brand of belt. Accurate judgment of quality depends upon wide experience in handling leather and a good all-round knowledge of the specific characteristics of the leather in different parts of the hide.

Matching. — After the strips have been sorted, they are sent to the *fitters* to be matched and have the laps marked. Pieces cut from the right side of the hide are matched with pieces cut from the left side, because all strips which are not backbone center pieces will stretch in a curve if subjected to a sufficient strain. Narrow strips from a properly stretched side, merely as a result of the stripping, contract to a slight curve. Belts made by joining alternately rights and lefts will roll out in a curve on the floor but will run true upon the pulleys, while belts in the construction of which no attention is paid to the matching of rights and lefts will roll out straight on the floor but will invariably stretch crooked if subjected to sufficient tension on the pulleys.

Marking the Laps. — The laps are marked according to the thickness of the stock and usually range between 4 inches and 10 inches in length. Laps must be longer on the shoulder end of the piece, because hides become thinner near the shoulder, and longer laps are needed, therefore, to maintain a uniform thickness of belt. Shoulder ends are joined to shoulder ends, and butt to butt, because the length of the laps match better, the thickness is more uniform, the stock is similar in quality, and the component parts will therefore wear and stretch more uniformly. The laps are next scarfed and prepared for cementing.

Cementing. — The usual cement employed to stick the laps has for its basis animal glue. Each manufacturer is likely to have his own formula calling for certain additions to the glue solution and particular methods of compounding, but they all are much alike. About 1905, the now indispensable waterproof cement was introduced. Laps

stuck with this are absolutely unaffected by water, either cold or hot. Whatever cement is used, the process of *sticking* is the same in its essentials. The surfaces of the leather to be joined are coated with the cement, put together in their final position, placed between the plates of a hydraulic press and subjected to heavy pressure. From the presses the belt goes to the inspector, and then to the stock-room.

Double and Triple Belting. — The preceding description applies particularly to single belting, but the processes are much the same if double or three-ply belting is to be made. A liberal quantity of stock which has been scarfed and the flesh side of which has been cleaned up with a scraper, so as to remove grease, loose flesh, etc. (this being the side which is cemented), is placed upon the fitter's bench. He matches these together on a smooth surface against wooden blocks which are his standard of thickness, the pieces being matched to secure as uniform thickness as possible and so that the laps of one ply come about halfway between the laps of the opposite ply.

Belt Speed and Belt Thickness. — In tests made by French engineers to determine the most favorable conditions under which belting should be run, it was found that the greatest efficiency of transmission was obtained when oak tanned belts ran at a speed of from 65 to 80 feet per second. What is termed "chrome-tanned" leather belts ran most favorably at about 100 feet per second. The most satisfactory working tension was from 575 to 850 pounds per square inch of section. The tests indicated that the thickness of the belt should be from $\frac{1}{20}$ to $\frac{1}{30}$ of the radius of the pulley. When chromium-treated belts are used, which are more elastic, a thickness of about $\frac{1}{15}$ of the radius is permissible. In fact, a chromium-treated leather belt, 0.4 inch thick, gave good results on a pulley $9\frac{3}{8}$ inches in diameter.

Narrow, thick belts are more desirable and work more satisfactorily than wide and thin belts. According to the experiments of F. W. Taylor, it is advisable to use double belts on pulleys 12 inches in diameter or larger; triple belts on pulleys 20 inches in diameter or larger; and quadruple belts on pulleys 30 inches in diameter or larger. If thin belts are operated at high speed, they tend to run in waves on the slack side and travel laterally, especially if there are sudden load changes. This waving and snapping wears the belt rapidly and can be practically eliminated by having the thickness in proper proportion to the width. The speed at which belting runs has comparatively little effect upon its life until the velocity is higher than from 2500 to 3000 feet per minute. The life is affected principally by the power transmitted, the method of fastening the ends, and the care of the belting.

It is safe and advisable to use a double belt on a pulley 12 inches in diameter, or larger. Light double belting runs steadily, with a minimum of "snap" or vibration, and does not twist out of place like single belting. It is successfully used for counter belts where shifters are used and where the work is not sufficiently hard to demand a heavy double belt; it is especially adapted for use on cone or flange pulleys, as it will keep its place and is less liable to turn over, and at the same time is pliable enough to hug the pulleys like a single belt. Double belting, light or heavy, is not recommended for twisted belts at high speed, nor for wood-working where belts come into contact with a large amount of chips or shavings, nor for places where much oil or water are liable to fall onto the belt.

Single leather belting up to about 3 inches in width is usually $\frac{5}{32}$ inch thick and weighs approximately 13 ounces to the square foot; single belting wider than 3 inches is about $\frac{3}{16}$ inch thick and weighs from 15 to 16 ounces per square foot. Light double belts, such as are used for driving dynamos and other high-speed machinery having pulleys of comparatively small diameters, are about $\frac{9}{32}$ inch thick and weigh about 21 ounces per square foot. Double belting, such as is used for main belts, usually weighs from 25 to 28 ounces per square foot. Belts as heavy as 30 ounces per square foot are frequently used and are commonly known as "heavy double." Large engine belts are often made with three thicknesses of leather.

Navy Department Belt Specifications. — All leather belting must be made from strictly No. 1 native packer steer hides, or their equal.

Tanning. — All hides to be tanned with white or chestnut oak by the slow process (6 or 8 months); chemical processes must not be used.

Currying. — Leather to be thoroughly curried by hand, and not to be stuffed or loaded for artificial weight. Leather must not crack open on the grain side when doubled strongly by hand with the grain side out.

Cutting. — Belting to be cut from the central part of the hide, not farther than 15 inches from backbone or more than 48 inches from the tail toward the shoulder. Belts 8 inches and over must be cut to include the backbone.

Stretching. — All leather to be stretched 6 inches in the lengthwise direction of the butt, and not to exceed 54 inches after stretching. Centers and sides to be stretched 6 inches separately; that is, all the side leather from which widths under 8 inches are to be cut, must be stretched after being removed from the backbone center section; center sections to be stretched in exactly the same size for which they are used.

Laps. — For single belts up to 6 inches, laps must not exceed 6 inches, or be less than $3\frac{1}{2}$ inches in length; for single belts above 6 inches, laps must not be more than 1 inch larger than width of belt. For double belts, laps must not exceed $5\frac{1}{2}$ inches, nor be less than $3\frac{1}{2}$ inches. No filling straps will be permitted.

Cement. — All laps to be held securely at every part, with best quality of belt cement, and when pulled apart shall show no resinous, vitreous, oily, or watery condition. Belting to be stretched again after manufacture.

Weights. — Belting to weigh for all sizes of single belts, 16 ounces to the square foot; double belts as follows: 1 to 2 inches, 26 ounces per square foot; $2\frac{1}{2}$ to 4 inches, 28 ounces per square foot; $4\frac{1}{2}$ to $5\frac{1}{2}$ inches, 30 ounces per square foot; 6 inches and above, 32 ounces per square foot.

Rawhide Lacing Leather. — Only hand-cut green slaughter hides of very best quality to be used. Rawhide laces to be in the following sizes: $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$, $\frac{1}{2}$, $\frac{5}{8}$, and $\frac{3}{4}$ inch. They must be cut lengthwise from the hide and have an average ultimate tensile strength not less than:

Width..........	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	inch.
Tensile strength..	95	125	155	165	180	205	230	pounds.

Lacing Belts. — There are many ways of fastening the ends of belts together. Frequently belts are made continuous by joining the ends with a cemented lap joint. The ends of most joints, however, are held together either by lacing or by means of some special fastening device. Some of the different methods of joining belt ends by rawhide lacing are illustrated in Figs. 2 to 4, inclusive. Nar-

row belts, or those having little power to transmit, are often laced together as shown at *A* in Fig. 2. As will be seen, there is only one row of holes in each belt. To begin with, the ends of the belt should be trimmed square and the holes should be punched exactly opposite one another in the two ends of the belt. The lacing is usually started in the center and in such a way that the lacing will not be crossed on that side of the belt which is to run next to the pulley. The hair or grain side of the leather should always run next to the pulley, because this side is harder and tends to crack if placed on the outside. When the grainside is next to the pulley, it is compressed as it passes over the pulley, especially if the latter is of small diameter, whereas the outer side is stretched.

To join the belt as shown at *A*, the lacing should be drawn halfway through one of the central holes, say, hole No. 1. The upper end is then started down through hole 2 and up through hole 3, then back through hole 2, up through hole 3, and then over to hole 4. The end is then drawn up through a hole just above hole 4 where it is fastened. In order to prevent this end from being withdrawn, an incision is made halfway through the lace which is then twisted, thus forming a barb for anchoring the end. In order to lace the opposite side of the belt, the remaining half of the lace is passed up through hole 4, then over to hole 5, up through hole 6, and so on; after the lacing is completed, this end is attached below hole 1, as the illustration shows. While the lacing operation for each side has been referred to separately, it is preferable to lace both sides at the same time unless the belt is very narrow.

Straight-stitch Lace. — The "straight-stitch" lace is sometimes applied to wide belts, as illustrated at *B* in Fig. 2. The method of starting the belt lacing depends upon whether the number of holes is odd or even. If the number is odd, begin as shown by the upper illustration at *C*. Both ends of the lacing are passed down through holes 3 and 8 from the grain side of the belt. Lace *a* is inserted up through hole 3, down through hole 8, up through 4, down through 9, up through 5, down through 10, up through 5, down through 10, up through 4, down through 9, and up through 3. This end of the lace is then fastened directly back of hole 3 in the usual manner. The other end *b* of the lace is inserted through hole 7, down through hole 2, and so on until it is finally brought up through hole 8 and fastened directly back of this hole. This method of lacing is, of course, the same regardless of the number of holes, provided that number is odd. When the width of the belt is such as to require an even number of holes, the lacing should be started as illustrated by the lower view at *C*. After the lace is inserted up through holes 3 and 6 from the flesh side, the end *a* is passed down through hole 7, up through hole 4, down through hole 8, up through 4, down through 8, up through 3, down through 7, up through 2, and is then fastened in an extra hole directly back of hole 2. The lacing is completed by passing end *b* down through hole 2, up through 5, down through 1, up through 5, down through 1, up through 6, down through 2, up through 7, and then fastening it directly back of hole 7.

Lacing Through Two Rows of Staggered Holes. — Most belts are laced through two rows of staggered holes, a very common method being illustrated in Fig. 3. The lacing is first passed through the middle holes 1 and 8 of the second row, after which it is shifted until the ends are of equal length. The lacing from hole 1 is then inserted down through hole 2, up through 3, down through 4, up through 5, down through 6, up through 7, down through 6, up through 7, down through 4, up through 5, down through 2, up through 3, down through 8, and then fastened in the usual way back of hole 8. The opposite side is, of course, laced in the same manner and the end of the lace is fastened in an extra hole beyond hole 1. As previously mentioned, both sides of the belt should be laced alternately instead of lacing one side before beginning the other, especially if the belt is quite wide. The number of holes and their positions for different belt widths are given in the accompanying table.

Fig. 2. The "Straight-stitch" Method of Lacing Belts

The Hinge-laced Joint. — It is desirable to so fasten the ends of a belt together that the joint is as flexible as possible, so that the joint will readily pass around a small pulley. The hinge method of lacing (illustrated in Fig. 4) is intended to give greater flexibility. Incidentally, this lacing is the same on both sides of the belt. Double rows of holes are punched in each end of the belt, the holes in the second row being exactly opposite those in the first. The holes should be at least $\frac{1}{2}$ inch from the end of the belt and should be spaced about $\frac{3}{4}$ inch apart. The diameters of the holes and the widths of the laces for different widths of belts would be practically as follows: For belts up to 5 inches in width, the holes should vary from $\frac{3}{16}$ to $\frac{1}{4}$ inch in diameter, and the width of the lace, from $\frac{1}{4}$ to $\frac{5}{16}$ inch. A lace of light weight should also be

used. For belts varying from 6 to 14 inches in width, the holes should be either $\frac{9}{32}$, $\frac{5}{16}$, or $\frac{3}{8}$ inch, according to the width. The lace should be $\frac{3}{8}$, $\frac{7}{16}$, or $\frac{1}{2}$ inch wide and of medium or heavy weight. For belts exceeding 14 inches in width, the holes should be $\frac{13}{32}$ inch in diameter, the lace $\frac{5}{8}$ or $\frac{3}{4}$ inch wide and of heavy stock. In any case, the hole should be just large enough to admit the lace, to avoid weakening the belt. Frequently the lace holes are made oval by punching two holes close together. When the holes are made in this way, the oblong opening thus formed should be parallel with the edge of the belt and not the end.

Belt Laces and Holes for Laced Joints

Width of Belt, Inches	Width of Lace, Inch	No. of Holes	Distance of Holes from End	
			First Row, Inches	Second Row, Inches
1 –1¾	¼	2 or 3	⅜	....
2 –2½	5/16	3	⅜	¾
2¾–3¼	5/16	5	½	1
3½–4½	⅜	5	⅝	1⅛
5	⅜	7	⅝	1⅛
6	⅜	9	¾	1¼
8	½	11	¾	1⅜
10	½	13	1	1¾
12	½	15	1	1¾
14	½	17	1¼	2

To begin lacing the hinge joint, put the lace through hole 5 and draw the ends even; then place the other end of the belt between the ends of the lace, as illustrated to the right in Fig. 4; next insert lace *a* up through hole 8, then down between the ends of the belt and up through hole 5, down through 2, up through 5 and down between the ends of the belt. Continue by inserting lace *b* down through hole 8, up through 11, down through 8, up between the belt ends, down through 4, up between the ends, down through 7, up between the ends, down through 4, up through 1, down through 4, up between the ends, down through 7, up through 10, down through 7, up between the ends, down through 4, up through 1. This end of the lace is now fastened back of hole 1. The lacing is then finished by inserting lace *a* up through hole 9, down between the belt ends, up through 6, down between the ends, up through 9, down through 12, up through 9, down between the ends, up through 6, down through 3, up through 6, down between the ends, up through 9, down through 12; this end is then fastened back of hole 12. This method of lacing may be applied for any odd number of holes, the method being to simply start with the center hole in every case. If the width of the belt is such as to require an even number of holes in each row, the lacing may be started in either of the two holes nearest the center.

Fig. 3. Method of Lacing requiring Double Row of Holes

Twisted Rawhide Belt Lacing. — While the ordinary form of flat rawhide lacing has been widely used, it has two drawbacks, one of which is that the holes punched in the belt are so large that its strength is materially reduced, and the other is that the bulkiness of the lacing at the joint has a tendency to cause the belt to jerk as the joint passes over the pulley. In order to overcome these objectionable features, the twisted rawhide belt lacing has been brought out. This lacing consists of rawhide which is twisted into a small cord. The advantages of this method of joining a belt are: that it is only necessary to punch small holes for the lacing, so that the strength of the belt is not materially reduced; the lacing is not bulky, so that it does not interfere with the smooth running of the belt on the pulley while the lacing is in contact with it; and that the tensile strength of the lacing is relatively high. In joining a belt with this form of lacing, it is claimed that the result approaches a cemented splice as closely as it is possible for a laced joint to do. The tendency for the belt to jerk or jump when the lacing comes into contact with the pulley is practically eliminated and this greatly reduces the strain on the belt, so that the cost of maintenance is materially reduced.

Fig. 4. Hinge-laced Joint for Belts

Tests on Twisted Rawhide Lacing. — Tests made at the Pratt Institute on belts laced with flat lacing and with twisted rawhide lacing showed the following results: A 3-inch single leather belt carefully laced with a $\frac{3}{8}$-inch flat rawhide belt lacing showed an opening of $\frac{1}{8}$ inch at the abutting ends when a tensile strength of 200 pounds was reached; at 760 pounds, the opening between the ends was $\frac{1}{2}$ inch; and, when a stress of 810 pounds was reached, the belt broke at the lacing holes. A 3-inch single leather belt (taken from the same section of belting), laced with twisted rawhide lacing showed an opening of $\frac{1}{16}$ inch at the abutting ends when a tensile stress of 900 pounds was reached; at a stress of 1460 pounds, the opening between the ends was $\frac{1}{8}$ inch; and when a stress of 1800 pounds was reached the strands of the lacing broke. None of the lacing holes had "pulled out" or broken, and, as a matter of fact, the belt could have been relaced through the same set of holes. As there is no metal in this lacing, there is no danger of accidents from projecting ends.

Metal Belt Fasteners. — A great many belts, especially of the smaller sizes, are held together by metal fastening devices which are made in many different forms. Some of these fasteners are sold in lengths which may be cut to suit the width of the belt. One common type is provided with prongs which are driven down into the belt ends and

clinched, the body of the fastener extending across the joint and forming what appears to be a steel lace. Another type is composed of separate sections which have prongs that are driven into each end of the belt; the outer ends are formed somewhat like a hinge and connected by a pin which makes a flexible joint. Other fasteners are in the form of flat metal hooks that engage narrow openings in each end of the belt. There are many other types of patented fastening devices designed to provide a rapid means of joining belts and, at the same time, form joints that are strong, flexible, and smooth or even.

Wire-clip Hinge Belt Joint. — The steel wire belt joint, shown in Fig. 5, consists of wire clips which are clinched

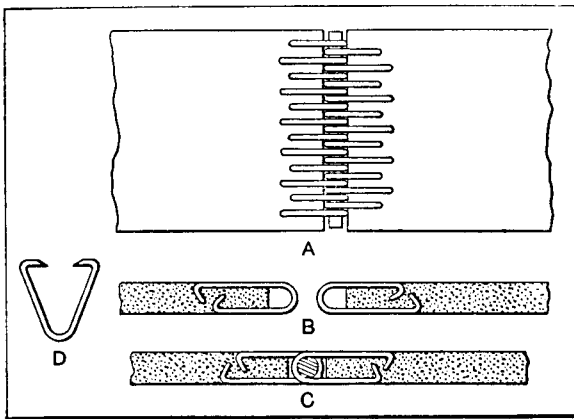



Fig. 5. Belt Ends joined by Wire Clips and a Rawhide Pin

in each end of the belt to be joined, forming loops at the outer ends and locked together by means of either a rawhide or bamboo pin. This method of lacing makes possible the disjoining of the belt without destroying the lace or in any way affecting the joining members, it simply being necessary to pull out the rawhide pin. The illustration shows two ends of a belt joined in this manner at *A*; at *B*, the pin has been removed, disjoining the belt; at *C*, a section of the belt is shown with a rawhide pin in place; and at *D* is shown the shape of the hook before being clinched in the belt. These hooks are clinched by means of a special tool. To facilitate the insertion of the hooks in the lacing tool, they are held in cards. For lacing a belt wider than the capacity of the lacing tool, the hooks are inserted in sections. On a belt of such a width as to require two lacings, one section is put in at one end, clinched, and brought flush with the belt, and another section of lacing is put in at the other end, thus completing the lacing of one end of the belt; the other end is handled in a similar manner. For a belt requiring three lacings, the hooks are first inserted in the center portion of the belt, after which the hooks are put in on both sides of the center portion.

Cemented Belt Joints. — The most satisfactory method of joining the ends of a belt is by making a cemented lap joint. The belt ends should be tapered to a smooth even surface, square with the edges, and the length of the lap should vary with the belt width approximately as given below:

Belt width in inches.........	1	2	3	4	5	6	7	8
Length of lap...............	5	5	6	6	7	8	8	9

For belting varying from 9 to 18 inches in width, the length of the lap is made equal to the belt width, and for widths above 18 inches, the lap need not be greater than 18 inches. Before cementing a joint, the belt should be placed over the pulleys and the proper tension obtained by means of clamps. These clamps are fastened across the belt on each side of the joint and are drawn together by threaded rods extending along the sides. Each clamp should be square with the belt and centrally located. After drawing the belt a little tighter than is desired, to allow for slack, bevel the ends to form the lap, and apply the cement to both surfaces while hot. If the belt is large, cement a few inches at a time, then rub with a "rub stone" and hammer down the cemented section thoroughly, proceeding in this way until the entire lap has been cemented. The joint should then be carefully hammered, especially along the edges. The clamps should remain in position for about an hour, or longer if convenient. Small and medium-sized belts are often joined by clamping the cemented lap between boards to which the belt has been tacked in the proper position.

Preparing the Belt. — In order to prepare an old belt that is greasy, the first step is to clean it thoroughly by soaking in benzine for about fifteen minutes. The belt is then laid out flat on a board or bench to allow it to dry. The ends that are to be joined should be further cleaned by placing blotting paper over and under them and then applying a hot iron to soften the grease so that it will be absorbed by the blotting paper. It is, of course, understood that a clean new belt does not require this treatment. In the case of a new belt, the belt should be fastened to a board or bench and the ends beveled as

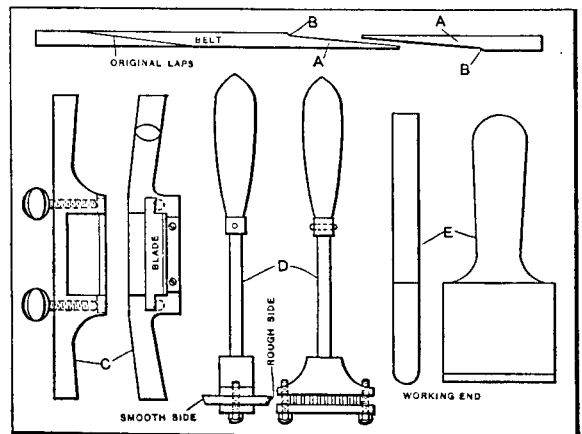



Fig. 6. Method of Forming Belt Laps, and Tools used for Forming and Cementing Operations

shown at *A*, the tools *C* and *D* (Fig. 6) being used for this purpose. The tool *C* is employed to rough down the ends to shape, and the smooth side of *D* is used to scrape the end of the lap to produce the shoulder *B*. The rough side of the tool is then brought into action to rough the surfaces to be joined so that they will take the cement. In beveling the ends of the belt, care should be taken to see that the finished lap will run in the same direction as the original lap in the belt, the length of the lap being determined from the length of the original lap of the old belt.

Belt Cements. — A good cement for joining the ends of a belt is the cheapest in the end. A good cement can be made from the following formula: Peter Cooper's white glue, 1 pound; powdered white lead, 1 ounce. This glue is heated in a double boiler and thoroughly stirred until it forms a thick paste. Before using, the paste is

thinned down the required amount with grain alcohol until a thin paste is formed. This paste is applied hot. After joining the ends of the belt, the joint is rubbed with the end of the wooden spatula *E*, taking care to apply pressure over the entire lap. Two small pieces of board can be clamped or nailed over the joint, after which the belt is placed in a warm, dry corner away from drafts. The joint should be allowed twenty-four hours to set before it is used.

The following preparation can also be used for cementing leather belts: Place equal parts of glue and isinglass in a glue pot. Add enough water to cover the two ingredients and let them soak ten hours. Then bring to boiling point and add pure tannin until the mixture appears like the white of an egg. Apply the cement while warm. For rubber belts, use 16 parts of gutta-percha, 4 parts of India rubber, 2 parts of caulker's pitch, and 1 part of linseed oil. These ingredients should be melted together and used while hot. This cement can also be applied to leather.

Isinglass for Belt Cements. — Isinglass is a term commonly confused with mica, although it is a totally different substance. Mica is a silicate mineral mined from the earth, while isinglass is an animal product made from the air-bladders of certain fishes. The best isinglass, which is used in superior belt cements, is made from the air-bladders of the Russian sturgeon caught in the Baltic Sea. This fish is the same as the sturgeon caught in American waters, but some peculiarity of the water or food in the Baltic makes a great difference in the quality of the air-bladder. The bladders taken from the American sturgeon are thin and of poor quality, being valued at about $1.50 per pound wholesale, while those taken from the Russian sturgeon are three or four times as thick and of far better quality, being valued at about $3.50 per pound. The difference is important to manufacturers and machine shop proprietors, as belt cement made with the best isinglass as its basis is stronger than leather. A belt properly joined with the best cement will break at some other point rather than at the cemented joint.

Inclination of Lap Joint. — There is a difference of opinion regarding the direction that the lap joint of a belt should incline relative to the direction of the motion of the belt. Some makers recommend the method shown at *A* in Fig. 7. As will be seen, the lap joint is so placed with relation to the direction of running of the belt that, when the belt passes over the pulley on which it is most likely to slip, there will be no tendency for the slipping pulley to pull the joint on the inside of the belt loose, and roll the scarfed end back, thus weakening the joint. When the loosened end rolls up in this manner, stresses are induced in the belt every time the joint passes over the pulleys, and these strains may be great enough to cause the belt to give way suddenly. In the illustration, the belt thickness is exaggerated in order to show the joint more clearly. The pulley on the left is the one most likely to slip, because of the smaller arc of contact with the belt; hence, the joint is so arranged that if this pulley slips, it will not roll back the scarfed end. A general rule based on this reasoning would be as follows: The leading point of the joint should be on the outside when the driven pulley is the smaller, but on the inside when the driving pulley is the smaller of the two.

The other method of inclining lap joints, which is sometimes recommended, is shown at *B*. For a single-ply belt, the leading end or point of the lap is on the pulley side. The lap is inclined in this way to prevent the end from opening, the theory being that when the leading end is on the outside, it tends to open up slightly, especially if the belt is operated at high speed, owing to the resistance of the air. As soon as there is a very slight opening, the atmospheric resistance tends to increase it, but when the leading end is next to the pulley, any tendency of the point to raise is overcome by frequent contact with the pulleys. At *C* is shown how the lap joints of a double or two-ply belt should be inclined. In this case, the lap of the outer ply is in the same relation to the direction of motion as for a single belt, but the lap of the inner ply inclines in the opposite direction. With this arrangement, the leading ends of the laps in both plies will be inside and protected, and the outer ends are to the rear and not subjected to the atmospheric resistance.

Belt Dressings. — Belts should be cleaned and greased every five or six months to give the grain side a soft adherent surface. The following mixtures are recommended: Take two parts of beef tallow to one part of cod liver oil (by weight); melt the tallow and allow it to cool until the finger can be inserted without burning; then add the cod liver oil and stir until cooled. A light coat of this mixture should be applied to the driving side of the belt after it has been cleaned. Rosin or rosinous mixtures should never be used to prevent belts from slipping. They will cause temporary adhesion, but the belt soon becomes glazed and slips more than before the rosin was applied. Lubricating oils should not be permitted to drop onto belts. If a belt has become saturated with oil, scrape it and pack it in dry sawdust or some other absorbent material for three or four days. When belting becomes dry, all surface dirt should first be removed before applying the dressing; this usually can be done by rubbing the belt with a cloth dampened with kerosene. If necessary, use a wooden or metal scraper. A dressing recommended for rubber belts consists of equal parts of red lead, black lead, French yellow, and litharge, mixed with boiled linseed oil and enough japan to make it dry quickly. Animal oil or grease should never be used on rubber belts.

Fig. 7. Inclination of Belt Lap-joint Relative to Direction of Motion

Causes of Belt Failures. — The causes of belt failures are many. In some instances, the cause may be traced direct to the manufacturer of the belt, who has not furnished belting of good quality; but often the best belting,

wrongly installed, will have need for frequent replacement. Under such conditions, the causes are two-fold: First, belt failures are often due to a defective working out of the underlying principles of practical belt transmission; and, second, belt failures are often the result of the selection of an improper type of belting for the particular kind of work for which it is to be used. A large percentage of the unnecessary costs of belting could be eliminated by the following suggestions: 1. By foreseeing and modifying the mechanical and architectural plans in the building of plants, so that they will conform to the most modern principles of practical belt transmission. 2. By modifying pulley conditions in a plant already built. 3. By the selection and proper installation of the right type and size of belting for the particular machine and pulley upon which the belt must operate. Very often, slight changes in transmission conditions would save great losses of time on many types of machinery in manufacturing plants. In the following paragraphs are given some of the common mistakes in transmission which could easily be rectified.

Short Centers Between Shafting. — Allowing too short centers between shafting often causes damage to the belt when the load is heavy. Owing to its limited length, the belt does not sag on the top side, as it would when driving from the bottom of the pulleys, which is the correct way; therefore, it becomes necessary to draw the belt very tight to avoid slippage, thus straining it and adding to the frictional resistance in the bearings. When the diameters of the driving and driven pulleys differ to any extent, as shown at *C*, in Fig. 12, there is a material reduction in the arc of contact on the smaller pulley, as the distance between the centers of the shafting is lessened, and, as the driving power of the belt, in a measure, depends upon the area of contact between the belt and pulley surfaces, the advantage of increasing the distance between the centers is apparent.

Installation of too Small Pulleys. — A slow running belt must be heavy if any strain is to be put upon it. It must also be of sufficient width to provide the proper area of contact to insure it against slippage. This is especially applicable in a case where small pulleys are used. If the pulleys, when installed, were enlarged proportionately, they would still give the same number of revolutions, but the belt speed would be increased, the grippage area increased, and a narrower and lighter belt could be used. Such a belt will invariably outlast a shorter, heavier, and thicker belt. In determining the minimum diameter of pulleys to be used with single and double belting, it is necessary to take into consideration the width of the belt to be used, the belt speed, and the horsepower transmitted. Whenever possible, the pulleys on which single belting is to run should not be made less than 8 inches in diameter, and those for double belting, not less than 12 inches in diameter.

Avoidance of Quarter-turn Drives. — It is a mistake to employ quarter-turn drives in belt transmission. This kind of drive can often be avoided by installing two loose pulleys so that the belt will operate over them as well as over the regular driving pulleys, as explained later in connection with sketch *A*, Fig. 12.

Avoidance of Small Pulleys on Long Center Drives. — Small pulleys should not be used on long center drives as they reduce the tractive power of a belt. Under such conditions, it is necessary that the belt be kept under a certain tension at all times. This usually results either in constant shortening in damp seasons or lengthening in dry periods.

Location of Driving Side of Belt. — It is important to so install a belt that the slack side is above and the driving side on the bottom of the pulleys. If this condition is reversed, the slack side being below, as shown at *B*, Fig. 12, the arc of contact of the belt with the pulleys is materially lessened.

Types of Belting for Different Classes of Work. — Perhaps the most common cause of wastefulness in belt transmission is the selection of improper types of belting for particular classes of work or machinery. In the following are given a few instances where practical mechanical knowledge of conditions should govern the selection of the belting. Drives with varying loads should be equipped with belts that have an abundance of elasticity and which are capable of retaining it. The elastic qualities should be such that, upon the relaxation of the load, the belt will return to its original state. Another instance where elasticity has great bearing upon the selection of the proper type of belting is in connection with quarter-turn belts. Here the belt must be able to form a long side and yet be elastic enough to prevent it from becoming permanent.

On drives where there is a long distance between centers, it is necessary that the belt be absolutely uniform and have perfect balance throughout. Unless a belt is procured which contains these qualities, its tractive powers and its driving powers will be materially reduced; its life will be short and it will be continually running off the pulleys. Only the best woven types of textile belts should be used on this character of drive, or leather belts made from absolute center stock. Often textile belts are selected for use with shifters. This should be avoided, however, unless roller shifters are used. If a belt is required to carry a heavy load, and it is necessary that the belt be kept tight at all times, one should be selected that is sufficiently heavy to insure a large factor of safety with regard to tensile strength.

Arc of Contact. — On drives where the arc of contact between the belt and the pulley is small, the transmitting efficiency of the belt will depend mostly on its width. Tensile strength, under these conditions, has very little to do with the horsepower the belt will transmit. The horsepower depends almost entirely upon the speed at which the belt travels. The area of contact of the belt with the pulley should, however, be large enough to prevent slippage.

Starting Machines with Full Load. — One mistake very often made by operators in large plants, which causes damage to the belting, is the starting up of machinery with the full load on. The belting is then required to transmit at a speed of, say, 100 feet per minute, what is probably the full amount of horsepower that it was designed to carry when running at a speed of, say, 2500 feet per minute. Unnecessary strains of this sort are among the prime factors which tend to destroy belting.

Effect of Water and Dampness. — One of the causes of the failure of belting is water and dampness. In order to avoid such a condition, the selection of the proper type of belting is of first importance. Dampness is, without doubt, the cause of more belt replacements than any other condition. While water itself may not splash on the belt, or the room in which the belt is operating may be free from water, nevertheless the action of dampness will often

injure the belting. Dampness, unless the leather is specially treated to withstand its action, will cause the very best of leather belting to stretch, run crooked, and, in many cases, open up the laps and piles. Belting at rest is affected much more by the action of moisture than when driving, as the frictional heat generated by the belt, when driving, tends to keep the belt dry. The waterproofing and damp-proofing of belting is of great importance at the present time. Many of the large manufacturing plants are now being constructed of reinforced concrete and cement blocks. However, unless special precautions are taken, it is almost impossible to keep a plant constructed in this manner free from dampness, especially in warm seasons when no heating system is in operation. Dampness will penetrate readily through concrete walls. In one instance, underground concrete conduits were built in which the driving pulleys were to operate, in order to save space by putting these belts under the floor, and also to get solid bearings for their shafting. Untreated, flat leather belting was used, but the dampness from the ground penetrated the concrete walls and made the place very unsuitable for belting which had not been specially waterproofed.

Belt Fasteners. — In determining the type of fasteners to use, consideration must first be given to the mechanical conditions under which the belt will operate. If the belt runs under a tightener or drives from both sides, then the joint must be of the hinge or flexible variety, or the belt be made endless. In the selection and application of fasteners, textile belting requires more care than does leather belting. The punching of a hole which is too large or too close to the end frequently results in the fastener tearing out. In punching a hole in leather belting, it is weakened only to the extent of the material removed, but the same operation on textile belting severs many of the interwoven strands and weakens the construction. The holes should, therefore, be made as small as possible and be placed at regular intervals. In preparing the ends of a belt for lacing or fastening, a square should be used so that the ends will be at right angles with the edges of the belt. Where fasteners do not give entire satisfaction, this condition is often due to the fact that the ends have not been cut true, consequently, when joined together, the strain is unevenly distributed across the width of the belt, falling more on some points than on others and resulting in the tearing out of the fasteners.

Effects of Belt Dressings. — In most belt dressings a certain amount of resin is used and in almost all dressings some form of graphite. While both of these compositions possess certain adhesive qualities, in time they are sure to injure the fiber of the leather. If leather belting is properly curried, it seldom becomes hard or dry, unless it is working under conditions where it is impractical to install a leather belt. Under such conditions it is advisable to use as a belt-dressing tallow mixed with a certain amount of castor oil. The tallow softens the fibers of the leather and the castor oil restores, to a large extent, the adhesive qualities in the belt, but even this most harmless type of dressing should be used sparingly, or the result will be a stretching of the leather. Where trouble is experienced through slippage of the belt, a few drops of castor oil on the pulley where the slipping occurs will be found to give good results. The oil will not injure the leather to the extent that most dressings will. The slippage of belts is generally due to the fact that frictional heat causes the grain or pulley side of the belting to become dry. Castor oil tends to soften the grain and reduce slippage. Belt dressings, if not used very sparingly, where there is a certain amount of dust in the air, will invariably clot on the belt and cause it to run unevenly.

General Service Conditions. — Most of the largest belt manufacturers have experimented with different types of belting on different classes of machinery and are able to give authoritative recommendations as to which type of belting will prove most economical for each machine. The problem of properly belting machinery is a large one, and new types of belting, new methods of fastening, and new types of belt dressings are being introduced frequently.

Rubber Belting. — Rubber belts are used in places exposed to the weather or the action of steam, as they do not absorb moisture or stretch as readily as leather belts, under like conditions. The quality of rubber belting depends on the mixture (containing more or less rubber) that forms the coating, the cotton duck that gives strength to the belt, and the method of manufacture. As to the rubber mixture, there are, in general, two kinds, one composed entirely of new rubber, and the other containing some, if not all, "re-worked" rubber. The latter is derived from discarded rubber articles such as rubber shoes, etc., and has lost much of its life. The best grades of rubber belting contain nothing but new rubber; the cheapest grades are composed largely of reclaimed rubber. The weight of the cotton duck is an important consideration. High-grade belts contain what is known as a 32-ounce cotton duck, and the cheaper grades have either a 30-ounce or 28-ounce duck. If the proper weight of duck is used, a 3- or 4-ply rubber belt is equal in strength to a single leather belt; a 5- or 6-ply rubber belt is equal to a double leather belt, and a 7- or 8-ply rubber belt is equal to a triple leather belt. A test commonly made to determine what is known as the "friction" of the belt, or the tenacity with which the different plies are held together by the rubber mixture, is as follows: The belt is cut so that the different plies of cotton duck can be pulled apart, and the amount of pull necessary to separate the plies determines the frictional value.

Rubber belting is ordinarily figured as averaging $\frac{1}{16}$ inch in thickness per ply; thus a 7-ply rubber belt is about $\frac{7}{16}$ inch thick, and corresponds in thickness to a heavy double leather belt. The permissible working load for average conditions may be taken as about 11 pounds per ply per inch of width; hence, a 7-ply rubber belt, 10 inches wide, should safely carry $7 \times 11 \times 10 = 770$ pounds maximum working tension. The coefficient of friction of rubber belting is about the same as that for leather belting, or, say, 0.3. Reuleaux limits the coefficient to 0.25. However, a still higher coefficient than 0.3 is frequently used for both leather and rubber belting in good condition and working under favorable circumstances.

According to the Boston Woven Hose & Rubber Co., when installing a rubber belt, the belt length should be made from $\frac{1}{8}$ to $\frac{1}{4}$ inch per foot shorter than the distance around the pulleys, the distance being measured with a steel tape line or wire drawn tightly; for a two- or three-ply rubber belt, the allowance should be $\frac{1}{4}$ inch to the foot; for a four- or five-ply belt, $\frac{3}{16}$ inch to the foot; and for a six-ply belt, $\frac{1}{8}$ inch to the foot. The belt should be stretched by means of belt clamps and it should never be applied to the pulleys by "running it on" after being

joined, but should either be laced together or spliced while in place.

Lacing or Splicing Rubber Belts. — Rubber belts are commonly joined together either by means of a rawhide lacing, a wire lacing, or by forming a lap and cementing the ends together. When a rawhide lace is used, the method of lacing illustrated in Fig. 3 may be employed. The first row of holes should be about $\frac{7}{8}$ inch from the end with the end holes $\frac{3}{4}$ inch from the sides of the belt. The second row should be about $1\frac{7}{8}$ inch from the end. These holes should not be less than $\frac{3}{4}$ inch apart and should be made as *small* as possible so as not to weaken the duck of the belt. For belts 12 inches wide or less, wire lacing is generally satisfactory. The holes should be made with a small brad awl and not with a belt punch, in order to avoid cutting the fabric. For a 4-inch belt, the holes should be about $\frac{1}{4}$ inch from the end of the belt, whereas for a 12-inch size, they should be about $\frac{7}{8}$ inch from the end, with other sizes in proportion. Belt hooks and clamps have proved satisfactory but should not be used where tighteners are employed or where the belts run over wooden pulleys.

When large belts are required, it is advisable to have them made endless. For belts larger than 12 inches, the step splice illustrated in Fig. 8 is recommended, which makes practically an endless belt. As will be seen, a lap joint is formed having three steps, three or four inches being allowed for each ply of duck. Care should be taken not to cut into the ply below when forming a step. A good quality of rubber cement should be applied to the prepared surface. Three coats should be used, each coat being allowed to dry before the next one is applied. The ends of the belt may then be firmly pressed together. The joint thus formed is reinforced either by means of sewing it with a light leather lacing or by inserting small copper rivets. When rivets are used, they should be placed about $1\frac{1}{2}$ inch apart and the rows should be staggered.

According to the New York Belting and Packing Co., when adjusting a new rubber belt, the allowances that should be made for stretching are as follows: The measurement should be taken around the pulleys with a steel tape drawn tightly, and, for a 3-, 4-, or 5-ply belt, $\frac{5}{32}$ inch per foot should be allowed for stretching. For a 6-, 7-, or 8-ply belt, allow $\frac{1}{8}$ inch per foot, and for 9- and 10-ply belts, $\frac{3}{32}$ inch per foot. Rawhide lace leather is recommended in preference to any other kind of lacing. The style of lacing recommended is similar to that illustrated in Fig. 3. When fasteners are used, they should be free from projections on the pulley side, and that style is preferred which does not have sharp angles on the prongs, in order to reduce the liability of tearing out. The diameters of the lace holes and their position for various widths of belts are given in the following table:

Belt Widths, Inches	Size of Hole, Inch	Size of Lace, Inch	Distance of First Row of Holes from Belt End	Distance of Side Holes from Belt Edge	Space between Holes, Inch
3	5/32	1/4	1/2	3/8	3/4
3 to 6	3/16	3/8	3/4	3/8	1 to 1 1/4
6 to 12	1/4	3/8	1	1/2	1 1/2
Over 12	5/16	1/2	1 1/4	1	1 3/4

Speed of Rubber Belting. — The speeds recommended for rubber belting are as follows: Minimum velocity of belt, 200 feet per minute; maximum velocity, from 5000 to 6000 feet per minute. No belt should have a speed of less than 200 feet per minute, and the best service is obtained with a speed of about 2000 feet per minute, although this speed may be increased to 5000 or 6000 feet, if necessary.

Pulleys for Rubber Belts. — One of the most common causes of failure in belting is the use of small pulleys which adds to the strain on the upper plies of the belt. Another source of trouble lies in the use of belts that are too narrow. The pulley should be as large in diameter as conditions will permit, thus increasing the arc of contact. The minimum pulley diameter recommended for a 4-ply belt is 12 inches; 5-ply belt, 18 inches; 6-ply belt, 24 inches; 7-ply belt, 30 inches; 8-ply belt, 36 inches; 9-ply belt, 42 inches; 10-ply belt, 48 inches. Larger pulleys are conducive to better results and will add to the life of the belts. An increase of power should be obtained, preferably, by using large pulleys, and wide belts rather than by an increase in the number of plies, in the belt, because the thinner belts are more flexible and have a better contact on the pulleys. The tensile strength of a 4-ply rubber belt of the best quality is given as 1000 pounds per inch of cross-section, or approximately 4000 pounds per

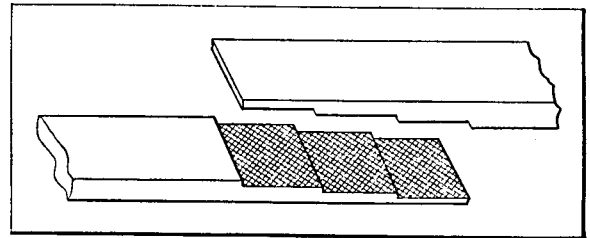
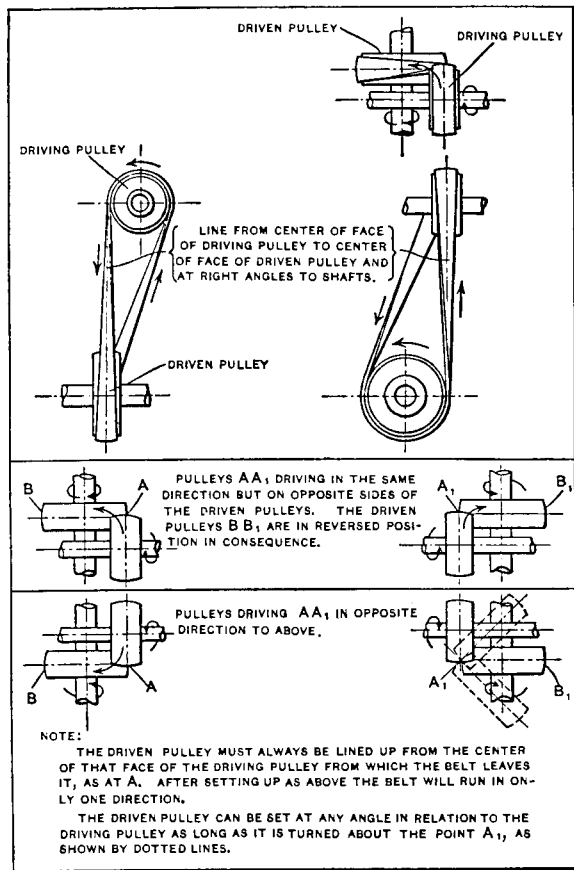



Fig. 8. Step Splice for a Rubber Belt

square inch. For economical and satisfactory service, a 4-ply rubber belt should not work under a greater strain than 65 pounds per inch of cross-section, and the tension on belting of other weights should be in proportion. A 12-inch, 4-ply belt weighs approximately $1\frac{5}{8}$ pound to the running foot, which may be used as a basis in computing the weights of other rubber belting.

Canvas Belting. — Canvas stitched belting is made of several laps or plies of cotton duck stitched lengthwise and the belt is afterwards treated with a compound made principally of linseed oil. This oil saturates the cotton duck, which is thus protected from dampness, and the belt is not easily injured by heat, cold, steam, gas, or acid fumes. Canvas stitched belting is often used where the material coming in contact with the belt or the surrounding atmosphere would ruin an ordinary leather, cotton, or rubber belt. It is applicable to belt conveyors, when the material to be handled will not cut the cotton fiber. For canvas belting, leather lacings are recommended for the smaller sizes, and step splices, like those used for rubber belts, for the larger sizes. Belt hooks and clamps are also used.

Fig. 10. Alignment of Pulleys for Quarter-turn Belt Drives

Quarter-turn Belt Drives. — When two pulleys are mounted on shafts located at right angles to each other and are connected by a belt, as illustrated in the upper part of Fig. 10, this is known as a *quarter-turn drive.* Obviously such drives should be avoided, if possible, because the belt is distorted as it twists around from one pulley to another, and, moreover, the contact between the belt and the pulleys is reduced, owing to the angular position of the belt. When installing a quarter-turn drive, it is important to align the pulleys in the proper manner. A general rule for aligning pulleys connecting shafts which are not parallel is as follows: The center of the face of the *driven* pulley must be aligned with the center of that face of the *driving* pulley from which the belt leaves.

Referring to the upper illustration in Fig. 10, it will be seen that the face of the lower or driven pulley is placed directly under that face of the driving pulley from which the belt leaves as it passes downward. The central illustration shows the upper driving pulleys A and A_1 rotating in the same direction but on opposite sides of the driven pulleys, in order to reverse the rotation of the latter. In each case, however, the driven pulley is aligned with that face of the driving pulley from which the belt leaves. Another modification is illustrated by the lower view; as indicated by the dotted lines at the right, the driven pulley may be set at any angle in relation to the driving pulley, provided it is turned about point A_1, where the belt leaves. The direction of rotation should not be reversed unless the relative positions of the pulleys are changed in accordance with the foregoing rule.

The operation of a quarter-turn belt can also be improved by installing it so that the stretching is equalized.

As the belt leaves the upper pulley, one side A (see Fig. 11) is loose and wrinkled to some extent, and all the work of driving is on side B, which is strained and stretched. The same thing occurs when the belt leaves the lower pulley. Cutting the belt on a radius and applying it so that the outside or long edge runs on that side where the stretching occurs is sometimes resorted to, but is objectionable in that it requires a special belt; moreover, it is difficult to determine just what curvature is required.

Where the quarter-turn drive cannot be avoided because of lack of space or for other reasons, the straining of the belt can be minimized by lacing the joining ends so that the flesh side of one and the hair side of the other is out. When a belt is placed over the pulleys in this way it is not twisted any more than with the common arrangement, for what was a quarter twist one way becomes a quarter twist in the other direction, when the belt end is turned halfway around. The advantage of connecting a quarter-turn belt in this way is that there is equal stretching of the belt on both sides, because a given point on one side of the belt is first on one side and then on the other side of the pulleys for each revolution, whereas, with the usual arrangement, all the stretching occurs on one side. Quarter-turn drives may be avoided in some cases by the use of a right-angle transmitter, or it may be preferable to install two idler guide pulleys on the top side of the belt.

Fig. 11. Method of Laying-out Floor Holes for a Quarter-turn Belt Drive

When a quarter-turn belt drive passes through an intervening floor, holes for the belt may be laid out by the following method. (See Fig. 11.) First draw an outline (plan view) of the two pulleys on the floor, directly below and above the respective pulleys to be connected by the belt. A starting point for this lay-out can readily be found with a plumb bob. Then draw the center-lines AB and CD through the faces of the pulleys and divide the diameter of each pulley into eight parts, as shown, numbering the divisions 1, 2, 3, etc. The numbers of the divisions must start from the sides of the pulleys which are opposite the arrow points (shown in the plan view) indicating the direction of rotation. Next, measure the distances from center to center of the shafts and from the center of the upper shaft to the floor. In the illustration, the distance from center to center of the shafts is 96 inches, and the distance from the center of the upper shaft to the floor is 42 inches. As $96 \div 8 = 12$, each division on the diameter of the pulleys is equivalent to 12 inches. Further, $42 \div 12 = 3\frac{1}{2}$, which is the number of spaces that the center of the belt will be from the center points of the sides of the upper pulley, as indicated at J and K in the engraving. Draw the line EJ through the point thus located in the rectangle representing the upper pulley. Then strike an arc with J as center and EJ as radius, as indicated, and divide it into eight equal parts. As the measurements from the floor to the upper shaft determine the number of spaces to be set off, these divisions should be numbered starting from point E, the line EJ being parallel to the face of the upper pulley; then set off $3\frac{1}{2}$ spaces from E, thus determining point L, and draw line LO through J, making JO equal to LJ. This line indicates the position of the center of the belt at the floor line; a line of the same length parallel to it through K indicates the other center-line of the belt at the floor line.

The lay-out for an angle of other than 90 degrees (see detail view, upper right-hand corner) differs only in that the arc on the pulley outline extends only from the line EJ to the line GJ, this latter line being parallel to the face of the lower pulley. Any number of divisions, either more or less than eight, may be used in the lay-out if more convenient.

Fig. 12. (A) Four-pulley Drive for Eliminating a Quarter-turn Drive. (B) Illustration showing Decreased Arc of Contact when Belt Drives on Top Side. (C) Extreme Variation in Pulley Diameters

The difficulty of driving with an ordinary quarter-turn belt drive depends largely upon the difference between the diameters of the driving and driven pulleys and the distance between them, the best results being obtained with narrow belts operating on comparatively small pulleys placed some distance apart. When the inclination of the belt relative to the pulleys is large, the belt is, of course, strained at the edges, and its arc of contact is reduced. Sometimes, in order to improve this condition, a guide pulley is placed against the slack side of the belt, so that the latter will pass over the pulleys in a more natural way.

Substitutes for Quarter-turn Belt Drives. — A good method of connecting shafts by belts which are at right angles to each other is illustrated at A in Fig. 12. This

form of drive, in general, is preferable to a quarter-turn drive. As the illustration shows, four pulleys are required, two being on the driving shaft and two on the driven shaft. These pulleys should be aligned with reference to one another as indicated by the plan view. One pulley on the driving shaft and one on the driven shaft are loose, these two pulleys merely acting as idlers. These loose pulleys revolve in a direction opposite the rotation of their shafts. The pulleys on each shaft should be of the same size. With this form of drive, the belt has a contact of 180 degrees for pulleys of the same size, thus providing a more efficient drive than the ordinary quarter-turn arrangement.

Another method of driving two shafts which are at an angle is illustrated at *A*, Fig. 13. This form of drive can

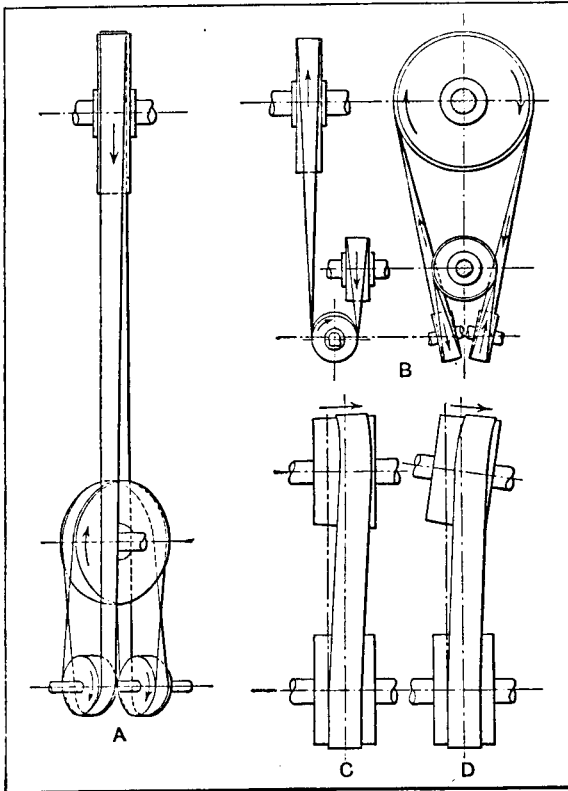



Fig. 13. (A) Angular Belt Drive. (B) Method of Connecting Closely-spaced Parallel Shafts. (C and D) Diagrams Illustrating Creeping Movement of Belts

be used where a quarter-turn belt would not be satisfactory. As will be seen, the belt passes from the driving pulley around idlers and then over the driven pulley. These idlers rotate in the same direction so that they may be fixed to a shaft which revolves in bearings, if this arrangement is preferred. The direction of rotation may be reversed with this form of drive, if necessary.

A method of connecting shafts by belts, when the shafts are at right angles and the lower pulley, for some reason, cannot be placed directly under the one on the upper shaft, is shown at *A*, Fig. 14. The belt is conveyed from the upper pulley horizontally to guide pulleys *G* and G_1, and then down to the driven pulley which, in this case, is on a shaft located beneath the floor. In order to obtain the best results with this method of driving, the length of the horizontal part of the belt should be longer than the vertical section. The arrangement when the driving or driven shafts are at right angles, but not in exactly the same plane, is illustrated at *B*. It will also be noted that there is considerable difference in the diameters of the driving and driven pulleys. The idlers over which the belt passes are commonly known as "mule" pulleys and the pulleys and their shaft, as a "mule stand." It will be noted that these pulleys are adjustable, thus making it possible to secure the proper alignment between the pulleys and the belt.

The view *C*, to the right, shows a similar form of drive, with the exception that, in this case, the driving and driven shafts are in the same horizontal plane and the driving and driven pulleys are of the same diameter. For guiding the belt, a plain non-adjustable mule stand is used or one having pulleys mounted on a vertical shaft.

Figs. 15 and 16 show two belt drives for machines standing at right angles to the line of shafting. By referring to Fig. 15, it will be seen that the belt passes over the driving pulley down to the pulleys on the machine (in some cases, one of these may be an idler); then the belt goes up to a "gallows" pulley that is suspended from the floor above. A modification of this same form of drive is shown in Fig. 16. In this case, the driven pulleys are above the floor, the belt passing up from the driving pulley around the driven pulleys, and then down to the gallows pulley. For some belt drives of this same general type, two smaller gallows pulleys are used instead of a single pulley, as shown in the illustrations. The distance between these smaller gallows pulleys is made to suit the drive.

Drive for Short Center Distance. — In some cases, two parallel shafts are placed so close together that they cannot be connected directly by a belt and transmit the required amount of power. A method of lengthening the belt, that is sometimes resorted to, is illustrated at *B*, Fig. 13. Two guide pulleys or idlers are used. When these pulleys are aligned as shown, the belt will operate in either direction. This form of drive is also employed when the driving and driven pulleys cannot be located in the same plane.

Creeping Movements of Belts. — Belts which are used for connecting parallel shafts have a tendency to run toward that part of the pulley which is largest in diameter; hence, pulleys are crowned to keep the belt in the center of the rim. If the shafts are not parallel and the pulleys are cylindrical, the belt will run toward the "low side" of the pulley or in the direction of the inclination of the pulley. The diagrams *C* and *D*, Fig. 13, illustrate the action of belts when operating under these different conditions. Diagram *C* shows two parallel shafts, one of which carries a cylindrical pulley and the other a conical pulley. While this combination would not be used in practice, it illustrates the principle involved. When pulleys are set in motion, the conical pulley tends to increase the speed of the right-hand side of the belt since the peripheral speed of the pulley at the right-hand edge is faster than where the pulley diameter is smaller. The result is that the belt leaves its normal path and gradually creeps over toward the right-hand side of the pulley, as indicated by the dotted lines. As previously mentioned, this tendency is taken advantage of by crowning pulleys or tapering the rim from the center toward each side. When the pulley is made in this way, each side of the belt tends to move toward the center of the rim and there is no trouble from the belt running off the pulley. Diagram *D* illustrates a case which is encountered in actual practice.

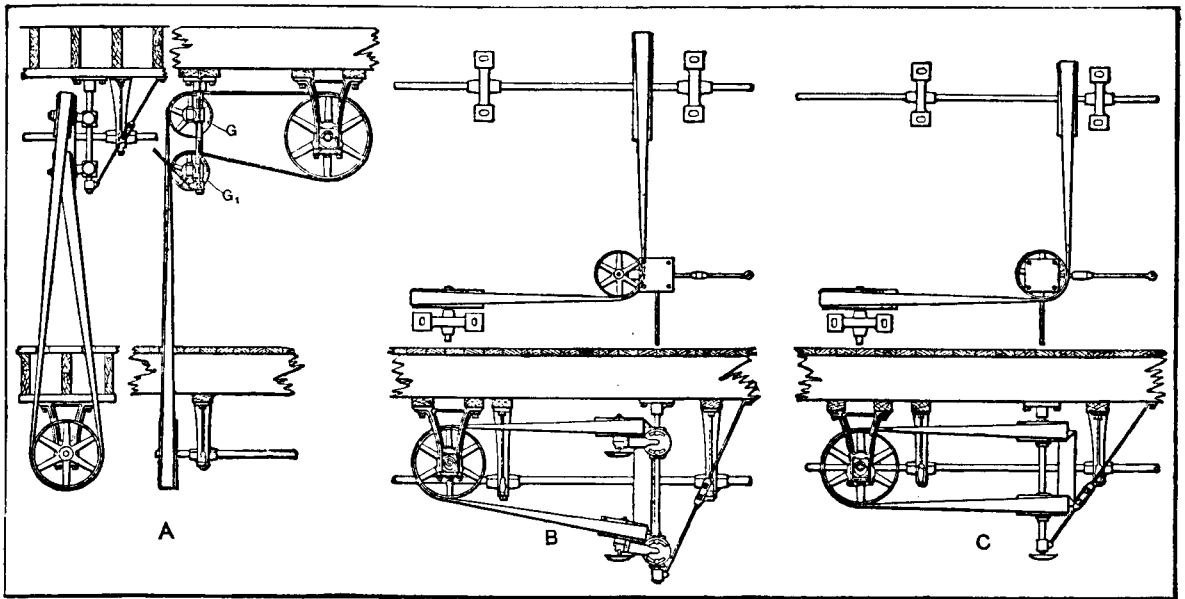



Fig. 14. Belt Drives for Shafts located at Right Angles

As will be seen, the pulleys are cylindrical but the shafts are not parallel. The result is that, as the belt passes onto the upper pulley, the right-hand side engages the pulley first, owing to the fact that the pulley is in an angular position, and the belt is carried over toward the low side, as the illustration shows.

High-speed Belt Drives. — When belts are run at very high speeds, there is often considerable slipping, especially with small pulleys such as are sometimes used to obtain the required speeds. This slipping and consequent loss of power may be the result of two causes: In the first place,

Fig. 15. Example of Right-angle Drive

when a belt is moving very rapidly, a thin layer of air tends to form between the belt and the pulley which destroys adhesion and prevents the transmission of a normal amount of power. In some cases, the trouble from this source has been so great that holes were drilled in the pulley face to provide air vents, thus preventing the formation of an air cushion between the belt and pulley. The action of centrifugal force on a high-speed belt also causes trouble as it tends to carry the belt out away from the pulley, thus decreasing the arc of contact and reducing the

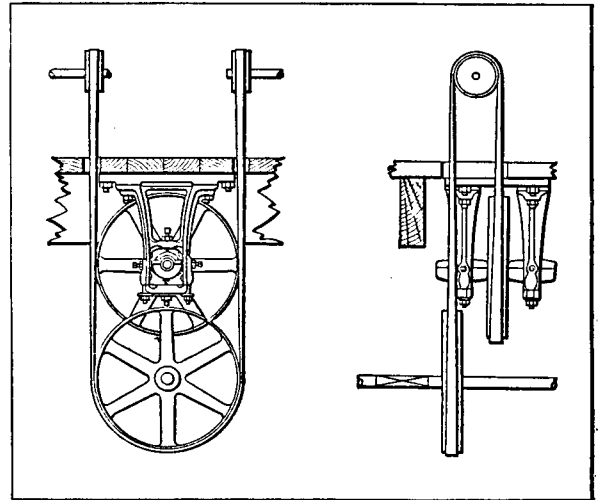



Fig. 16. Another Example of Belt Transmission between Shafts at Right Angles

amount of power transmitted. One method of overcoming this difficulty is to place an idler pulley on each side of the small driven pulley and a short distance from it. These idlers are mounted on pivoted arms held the required distance apart by turnbuckles, so that the two idlers can be drawn toward each other as much as may be necessary. This simple arrangement enables the belt to maintain a good grip upon the pulley, even when operating at high speeds.

Belt Lengths. — A simple rule for finding the length of a belt when both pulleys are about the same size, is as follows: Multiply the sum of the pulley diameters in feet by 1.57 and add to the product twice the distance between the shaft centers. When the size of one of the pulleys has

been changed, the change required in the length of the belt may be found by multiplying the difference in the diameters of the old and new pulleys in feet by 1.57; the product will be the number of feet necessary to remove from or insert in the belt, depending upon whether the new pulley is larger or smaller than the size previously used.

Pulley Diameters for Given Speeds. — To find the diameter of the driving pulley necessary to obtain any speed required for the driven pulley, multiply the diameter of the driven pulley by the number of revolutions of the driven shaft, and divide by the number of revolutions of the driving shaft; the result will be the diameter of the driving pulley. Similarly, to find the diameter of a driven pulley necessary for a given number of revolutions, multiply the revolutions of the driving shaft by the diameter of the driving pulley and divide by the required number of revolutions of the driven shaft.

Arc of Belt Contact. — When two pulleys *A* and *B*, Fig. 17, are of equal size and connected by an open belt, the arc of contact *a* will be approximately 180 degrees (in actual practice the contact exceeds 180 degrees, owing to the sag of the belt). When the drive from pulley *A* is to a smaller pulley *C*, obviously the arc of contact *x* is reduced, the amount of reduction depending upon the difference in the pulley diameters and their center-to-center distance.

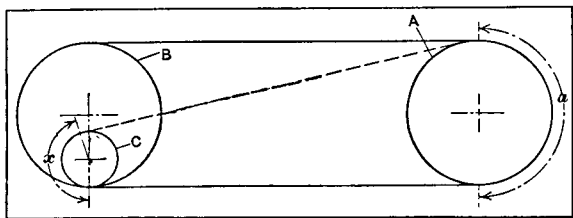



Fig. 17. Variation in Arc of Belt Contact caused by Change in Size of Driving or Driven Pulley

As the effective pull or difference in tension between the tight and slack sides of the belt is reduced as the arc of contact diminishes, this affects the amount of power that a given belt will transmit. For an open belt drive without idler pulleys, the arc of contact in degrees for the smaller pulley may be found as follows: Multiply the difference between the diameters of the pulleys in inches by 4.75; divide the product by the distance between the pulley centers in feet, and subtract the quotient from 180. Assuming that pulley *A* is 40 inches in diameter, pulley *B*, 10 inches, and the center-to-center distance, 8 feet, the arc of contact would equal:

$$180 - \frac{(40 - 10) \times 4.75}{8} = 162.2 \text{ degrees.}$$

Leather-covered Pulleys. — As the friction of leather on leather is much greater than of leather on iron or steel, leather-covered pulleys will transmit considerably more power than plain uncovered pulleys for the same belt tension. Before covering a pulley, clean the surface of all grease by washing it with naphtha or gasoline. Make the cover endless and about ⅛ inch to the foot shorter than the circumference of the pulley. Then place the endless cover on the pulley for a distance of about one inch. Next cover with glue the exposed inside surface of the cover and the exposed surface of the pulley. After applying the glue, drive the cover on by lifting the pulley and striking the cover edges against the floor or bench. This should be done quickly and by striking lightly to prevent bending the cover. When the cover is in place, rub the edges with a round stick or handle to secure a good contact. A properly covered pulley does not need rivets, although it is customary to insert a few copper rivets. Pulleys should be allowed to set two or three hours before using. Covering pulleys with leather reduces the slipping of belts and decreases the friction loss, thus making it possible to run the belts with less tension, which increases the life of the belt and reduces the friction on the bearings. It is estimated that leather-covered pulleys will enable belting to transmit 30 per cent more power than pulleys having a smooth iron surface.

Crowning of the Face of Pulleys. — The amount of "crowning" that should be given to a pulley differs with the conditions under which it works. The amount should be greater for leather belting than for cotton belting and also greater for low speeds than for high speeds. Different authorities recommend very different amounts of crown. One recommends a crown of (or height at center) $\frac{1}{20}$ of the width of the pulley in the case of leather belting and $\frac{1}{150}$ of the width for cotton belting. Another recommends from $\frac{1}{16}$ to $\frac{1}{8}$ inch per foot width of crown for high speeds and $\frac{1}{4}$ inch for low speeds, but these figures must be modified to meet individual conditions. The crowning of a pulley tends to keep the belt on only when the belt as a whole does not slip. A slipping belt will run off a crowned-face pulley quicker than from a straight-faced one.

Mechanical Belt Shifters. — When belt-driven machine tools such as lathes, milling machines, etc., are equipped with cone pulleys, the shifting of the belt from one step of the pulley to another is somewhat dangerous when done by hand and may be quite difficult, especially when the overhead countershaft is comparatively high and the belt is under considerable tension. In order to facilitate the changing of belts on cone pulleys, mechanical shifting appliances have been devised. A simple form of mechanical shifter consists of a guard which surrounds the belt and is mounted on a horizontal rod extending parallel with the countershaft. As the belt passes through the guard, a horizontal movement of the latter, effected by means of a shifting lever, forces the belt off of whatever step of the cone pulley it happens to be on.

A mechanical shifter which has more complete control of the belt consists of two racks, one of which is mounted on the headstock of the machine and the other on the countershaft. A belt shifter is secured to each rack and, in order to shift the belt from one step on the cone pulley to an adjacent step in either direction, it is merely necessary to turn a crank handle through one complete revolution. This crank operates two pinions that are in mesh with the racks, the connection being through bevel gears and a telescopic shaft, the length of which may be adjusted for different heights of ceilings.

The two racks and belt shifters are fixed relatively to each other and one shifter acts one-half revolution of the crank in advance of the other. In this way, one belt shifter throws the belt from one step on the machine cone to the next smaller step by turning the crank through one-half revolution; the other belt shifter then comes into action and the remaining half revolution shifts the belt on the countershaft cone pulley from one step to the next larger step. If a reduction of speed is desired, the crank is turned in the reverse direction and the steps just referred to will then be reversed. As the crank is always turned through one complete revolution, the operator becomes

accustomed to reaching for the handle in the position that it always assumes. The operation of this belt shifting device is such that the belt is automatically locked in position and cannot be shifted except by turning the crank. Stops are also provided in both directions which make it impossible to throw the belt entirely off the cone pulleys. It is stated that belts are longer lived where this shifting device is used because they are held in place and cannot curl up against the edges of the adjacent step of the cone pulley. For this reason, wider belts can also be used on a cone pulley with steps of a given width.

Horsepower Transmitted by Belting. — The horsepower which a belt of given size should transmit, depends principally upon the speed of the belt and the working stress or pull per inch of width to which it should be subjected. There is considerable difference of opinion regarding the proper working stress, owing to the fact that experiments have not been made upon the same basis. If the problem is to determine the maximum amount of power a given belt will transmit, naturally the working stress will be higher than when the durability and the cost of repairs are considered as well as the power to be transmitted. The approximate number of horsepower that may be transmitted by a belt can be determined by simply multiplying the *effective pull* in pounds per inch of belt width by the width of the belt in inches and the speed of the belt in feet per minute, and dividing the product thus obtained by 33,000. The effective pull is the difference in tension between the tight and slack sides of the belt. A commonly used value for the effective pull is 35 pounds per inch of width for single belts, and from 55 to 65 pounds for double belts. Extensive tests conducted by Fred W. Taylor indicate that these values are excessive when the life of the belt and the expense incident to belt failures are considered, and that a pull of 35 pounds per inch width for oak-tanned and fulled double belts will give the most satisfactory results. The most economical speed for belting was found to be between 4000 and 4500 feet per minute. In the formulas:

D = diameter of driving pulley in inches;
V = velocity of belt in feet per minute;
N = number of revolutions of pulley per minute;
S = effective pull of belt per inch of width, in pounds;
W = width of belt in inches.

Then:

$$V = \frac{\pi DN}{12} = 0.2618\,DN; \quad \text{H.P.} = \frac{SVW}{33{,}000} = \frac{0.2618\,SDNW}{33{,}000}.$$

As the effective pull is an assumed quantity of uncertain value, it is, of course, not necessary to retain in the formula so exact a quantity as 0.2618. If this number is given in round figures as 0.25 or ¼, the formula is simplified as follows:

$$\text{H.P.} = \frac{SDNW}{4 \times 33{,}000}.$$

Example: — What horsepower can be transmitted by a 2½-inch belt, assuming that the driving pulley is 12 inches in diameter, the speed of pulley 200 revolutions per minute, and the effective pull 33 pounds per inch of width? Inserting these values in the formula:

$$\text{H.P.} = \frac{33 \times 12 \times 200 \times 2.5}{4 \times 33{,}000} = 1.5.$$

If the horsepower to be transmitted is known, the required width of belt can be found by transposing the given formula as follows:

$$W = \frac{\text{H.P.} \times 33{,}000 \times 4}{SDN}.$$

A simple rule for determining approximately the horsepower that should be transmitted by rubber belting is as follows: Multiply the number of plies in the belt, its width in inches, the pulley diameter in inches, and the speed in revolutions per minute; then divide this product by 12,000. Expressing this rule as a formula:

$$\text{H.P.} = PWDN \div 12{,}000,$$

in which

H.P. = horsepower;
P = number of plies in the belt;
W = width of belt in inches;
D = diameter of pulley in inches;
N = revolutions of pulley per minute.

The problems in belting, although apparently simple, are in reality, so complicated, and so many considerations are involved in their solution, that there is considerable variation in the rules and formulas for determining the power transmitted by a given belt. There are, in general, five principal considerations which should be taken into account in determining the size of belt, *viz.*, coefficient of friction, arc of contact, speed of belt, strength of belt per square inch of section, and durability. Experimental work has usually been directed to determine the coefficient of friction, and it has been shown by exhaustive tests that the coefficient of friction of leather belting on smooth cast-iron pulleys may vary from 12 to 165 per cent, depending upon so many factors that it is impossible to so correlate them as to determine accurately what the coefficient may be in any given case. In general, the value of the coefficient depends upon the nature and condition of the leather, the percentage of slip, temperature and humidity, and probably pressure.

The slip on the pulleys, which influences the friction

Horsepower Transmitted by Leather Belting

The body of the table below gives the value of F in the equations.

$$\text{H.P.} = \frac{V \times W}{F} \quad \text{and} \quad W = \frac{\text{H.P.} \times F}{V},$$

in which H.P. = horsepower transmitted; V = belt velocity in feet per minute; W = width of belt in inches.

Example: — How wide should a single belt be in order to transmit 2 H.P. at 600 feet per minute over a 4-inch pulley with 140 degrees wrap?

From the table the value of F for the given conditions is found to be 1270. Inserting this and the known values in the formula for belt width:

$$W = \frac{2 \times 1270}{600} = 4.23 \text{ inches.}$$

Belt Thickness	Diameter Small Pulley, Inches	Arc of Contact							
		210°	200°	190°	180°	170°	160°	150°	140°
Single	Up to 8	1010	1040	1070	1100	1140	1180	1220	1270
Single	8 to 36	830	860	890	920	950	990	1040	1100
Single	Over 36	750	770	800	830	860	890	930	980
Double	Up to 14	570	590	610	630	650	670	700	730
Double	14 to 60	470	480	500	520	540	570	600	630
Double	Over 60	430	440	450	470	490	510	530	560
Triple	Up to 21	400	410	420	440	460	480	500	520
Triple	21 to 84	330	340	350	370	390	410	430	450
Triple	Over 84	300	310	320	330	340	360	380	400

more than any other factor, is directly proportional to the belt speed, and may safely be estimated at one per cent of the speed per pulley; that is, two per cent for each pair of pulleys. The maximum slip permissible with horizontal belts is about 20 per cent; a greater slip than this is liable to throw the belt off.

For many years a coefficient of about 0.42 was generally accepted by engineers; this was deduced from the results of experiments having a large slip (200 feet per minute), which gave the coefficient equal to 0.58, so that the value recommended is only 0.70 of that obtained by experiment. Even with this allowance it is probable that the value is somewhat higher than average practice would warrant. With percentages of slip varying from 0.8 to 2 per cent, Lewis finds the coefficient to vary from 0.25 to 0.38, the first value being obtained for a percentage of slip equal to 1.5, and the second for a slip of 1.7 per cent. Two per cent slip gave a frictional coefficient equal to 0.45. Other experimenters find that the coefficient varies from 0.12 to 0.58, depending principally upon the rate of slip. Numerous tests made at the Massachusetts Institute of Technology indicate that 0.27 is the most suitable value to use for a low rate of slip. These results were obtained on belts running about 1500 feet per minute and with a slip of about three feet per minute.

Regarding the strength of belting, this is extremely variable. Kirkaldy's tests, which cover a wide range, show the strength to vary from 2000 to 6000 pounds per square inch of section. The various tests indicate that double belts are superior to single, both in respect to their greater uniformity and also a lesser liability to become crooked by reason of uneven stretching, probably due to the fact that the former are composed of two layers, which tend to neutralize any local imperfections occurring in either. It is evident that the strength of a belt cannot be stronger than that of the joint. With cemented joints, the splice, if properly made, is not materially weaker than the body of the belt; with riveted joints, the ultimate strength, as given by Towne, is 1750 pounds per square inch, and for laced belts, 960 pounds. Laced joints may rupture under a stress of from 30 to 60 per cent of the strength of a solid belt. In existing belt transmissions, the stress per square inch of section is extremely variable. The reasons for this are two-fold. First, a superior quality of leather will permit a greater stress without undue trouble and expense in repairs; and, second, some belts are calculated to transmit a stress greatly in excess of that to which they are actually subjected.

Horsepower Transmitted by Leather Belting *

Table giving number of horsepower transmitted by belts one inch wide, considering the effects of centrifugal force, so that the tension on belt is constant at all speeds.

Speed in Feet per Minute	Thickness of Belt				Speed in Feet per Minute	Thickness of Belt			
	Single	Double	Triple	Four-ply		Single	Double	Triple	Four-ply
100	0.14	0.24	0.33	0.44	3400	3.89	6.74	9.10	11.96
200	0.27	0.48	0.67	0.88	3600	4.03	6.95	9.35	12.28
300	0.41	0.73	1.00	1.32	3800	4.14	7.12	9.55	12.57
400	0.54	0.96	1.33	1.75	4000	4.24	7.26	9.70	12.73
500	0.68	1.21	1.66	2.19	4200	4.33	7.36	9.79	12.84
600	0.81	1.44	1.99	2.62	4400	4.39	7.42	9.83	12.88
700	0.95	1.68	2.31	3.05	4600	4.43	7.44	9.80	12.84
800	1.08	1.93	2.64	3.48	4800	4.45	7.42	9.72	12.71
900	1.21	2.15	2.96	3.90	5000	4.45	7.37	9.56	12.50
1000	1.34	2.38	3.28	4.32	5200	4.43	7.26	9.34	12.20
1100	1.47	2.61	3.59	4.73	5400	4.38	7.10	9.05	11.80
1200	1.60	2.85	3.90	5.14	5600	4.31	6.92	8.69	11.30
1300	1.73	3.07	4.21	5.55	5800	4.21	6.65	8.25	10.70
1400	1.86	3.30	4.51	5.94	6000	4.09	6.35	7.73	10.00
1500	1.98	3.53	4.81	6.34	6200	3.94	6.01	7.13	9.19
1600	2.10	3.73	5.10	6.72	6400	3.76	5.58	6.44	8.26
1700	2.23	3.94	5.39	7.10	6600	3.56	5.11	5.67	7.22
1800	2.34	4.15	5.67	7.47	6800	3.32	4.57	4.80	6.06
1900	2.46	4.35	5.94	7.83	7000	3.05	3.98	3.84	4.77
2000	2.58	4.56	6.21	8.18	7200	2.75	3.31	2.79	3.36
2200	2.80	4.94	6.73	8.85	7400	2.42	2.60	1.64	1.82
2400	3.01	5.30	7.21	9.51	7600	2.05	1.82	0.39	0.14
2600	3.21	5.65	7.67	10.09	7800	1.65	0.95	.....	.....
2800	3.40	5.97	8.09	10.64	8000	1.21	.....	.....	.....
3000	3.58	6.25	8.47	11.14	8200	0.74	.....	.....	.....
3200	3.74	6.52	8.80	11.58	8400	0.23	.....	.....	.....

In all the above data it is assumed that the arc of contact of the belt is not less than 180 degrees.

If this arc is.................	90°	112½°	120°	135°	150°	157½°
Divide H.P. given by.......	2.21	1.72	1.6	1.4	1.24	1.17

* Note. — This table is based on an effective pull of 45 pounds per inch of width for single belts 3/16 inch thick; 80 pounds for double belts 3/8 inch thick; 110 pounds for triple belts 9/16 inch thick; and 145 pounds for 4-ply belts 3/4 inch thick. The table shows that there is no advantage in running belts faster than from 4400 to 4800 feet per minute, due to the action of the centrifugal force.

Durability of Belts. — A feature often neglected in the ordinary formulas for determining the horsepower that may be transmitted by belting is that of durability of the belts. It has been recommended that the one consideration which should have more weight than all others in making up tables and rules for the use and care of belting is how to secure the least possible interruption to manufacture from this source. In calculating the total expense of belting and the manufacturing cost chargeable to this account, the largest item is the time lost on the machines while the belts are being relaced and repaired, especially in those plants where the running of one series of machines is dependent one upon another.

Some of the results of Taylor's observations on belting, covering a period of nine years, are given in the following:

1. The belt speed for maximum economy should be from 4000 to 4500 feet per minute.

2. Belts are more durable and work more satisfactorily when made narrow and thick rather than wide and thin.

3. It is advisable to use double belts on pulleys 12 inches in diameter or larger, and triple belts on pulleys 20 inches in diameter or larger.

4. When a belt is spliced, the tension used should be ascertained by means of a spring balance connected to belt clamps.

5. The total life of belting, cost of maintenance and repairs, and the interruption to manufacture caused by belts are dependent upon the total load to which the belts are subjected more than upon any other condition. The other conditions are: Method of splicing, care in properly greasing and keeping them clean and free from machine

oil, and the speed at which they run. This latter has little effect under 2500 feet per minute.

6. The total stretch of leather belting exceeds six per cent of its original length.

7. Oak-tanned and fulled belts are superior in all respects (except coefficient of friction) to either the oak-tanned not fulled, the semi-rawhide or rawhide with tanned face.

8. If double leather belts are tightened while at rest, 71 pounds per inch of width, and subjected to an additional working load of 65 pounds per inch of width, their tension will fall in two and one-half months so as to be, while at rest, 33 pounds per inch of width, or 106 pounds per square inch section; their average tension during these two and one-half months being 46 pounds per square inch of width, or 150 pounds per square inch section; their average load during these two and one-half months being 111 pounds per inch of width, or 358 pounds per square inch of section. These are the conditions under which belts work when tightened according to the ordinary rules.

9. The most economical average total load for double belting is from 65 to 73 pounds per inch of width; *i.e.*, from 200 to 225 pounds per square inch of section.

10. A double belt having an arc of contact of 180 degrees will give an effective pull of 35 pounds per inch of width for an oak-tanned fulled belt, and 30 pounds for other leather belts and six to seven-ply rubber belts; or 950 feet per minute of one inch wide oak-tanned and fulled belt will transmit one horsepower, and 1100 feet per minute of other leather belting and six- to seven-ply rubber.

11. Belts should be cleaned and greased every five or six months.

While the foregoing conclusions are valuable from an engineering as well as commercial standpoint, the first cost of equipping a plant under these rules would often prohibit the use of such heavy belts; moreover, in many situations it is probable that equally satisfactory results are obtained under stresses twice as great as those given, especially in those cases where the belt runs at moderate speed over a pair of approximately equal pulleys of large diameter, and is not shifted from one pulley to another. On the other hand, the conditions may require certain belts fully as wide if not wider than those advocated by Mr. Taylor. Such a case might arise when running at a high belt speed from a large to a very small pulley (or *vice versa*) with the tight side of the belt on top; in this case, the centrifugal force would diminish the effective tension, the arc of contact would be lessened, and the adhesion of the thick belt on the small pulley would not be as effective owing to imperfect contact, due both to air entrainment and to a wrinkled belt produced by bending; all of these would necessitate an increased width of belt.

Owing to these various causes, it is impracticable to determine a general formula which will be applicable in all cases. Usually the conditions are such that the principal modifying agencies are known, and, if these be taken into account by choosing suitable constants (to be selected for any particular case when the data are known), a proper belt width may be obtained which will be entirely satisfactory as regards durability and expense of operation, in ordinary practice.

There are, in common use, two radically different methods of making belt connections between a pair of shafts for the transmission of power between them. By one method, a belt is run directly from a pulley on one shaft to its mating pulley on the other without the use of idle, guide, or tightening pulleys, thus obtaining what are termed "straight stretches" of belt between the working pulleys. By the other method, idle pulleys are introduced to increase the angle of wrap of the belt about the pulleys. One or more such "idlers" may be used, according to the nature of the drive, and, in any case, one of the idlers is used for a belt tightener. In the majority of belt drives with a large angle of wrap, the belt speed is higher than is commonly used with straight stretches of belt between the working pulleys. This does not apply, however, where the belt runs from an engine, because the peripheral speed of an engine pulley is often so high at the normal engine speed as to make it inadvisable to increase it for any purpose. There is no reason why the belt speed should be in any way affected by the angle of wrap; but it is more convenient to place large diameter pulleys when guide pulleys are used than when they are not, for the belt is more readily kept out of the way when idlers are used to guide it. Hence, for a given rate of rotation, a system with guide pulleys and large angles of wrap affords a more convenient means of securing high belt speed than straight stretches of belt between the driving and driven pulleys.

Losses at the Bearings. — A considerable amount of the power loss in a belt transmission system is at the journal bearings supporting the parts. The pressure against the bearings is partly due to the belt tensions and partly to the weight of the parts supported by the bearings. If the belt tensions can be kept low, without increasing the weight of the parts supported by the bearings, there will be a reduction of the power lost at the bearings. This reduction is on account of the reduced pressure on the bearings. The belt tensions can be reduced by increasing the belt speed, within certain limits, in a system for transmitting a given amount of power. This increased belt speed can be secured by using larger pulleys without changing the rate of rotation per minute. The centrifugal action on the belt is the chief factor that limits the belt speed. There are exceptional cases where decreasing the belt tension by increasing its speed will not reduce the pressure on the bearings to any great extent.

Effect of Belt Pull in a Vertical Drive. — As an illustration, consider a drive where a belt connects two pulleys of equal diameters on parallel shafts, one directly above the other. If the weight of the lower pulley and of the part of the shaft supported by the bearings is 200 pounds (which would, of course, cause a total of 200 pounds pressure on its bearings, if they are on opposite sides of the pulley) and the belt tensions are such that the belt just lifts the weight of the pulley and shaft, there may be no pressure on the bearings of the lower pulley. Assume that, by increasing the speed of the belt, its lifting action on the lower pulley is reduced to 150 pounds. If the weight of the lower pulley and its shaft remains 200 pounds as before, there will be a downward pressure of 50 pounds on the bearings instead of nothing, as before. At the same time, if the weight of the parts supported by the bearings of the upper pulley remains unchanged, there will be a reduction of the pressure on its bearings by 50 pounds on account of the reduced downward pull of the belt. On the whole, therefore, the total sum of the bearing pressures will be the same in both cases, and there will be no reduction of power loss at the bearings.

The case just mentioned is an exceptional one, because, in mechanical power transmission machinery, the sum of

the tensions in the stretches of belt leading off vertically upward from a pulley, or nearly so, are almost invariably great enough to more than overcome the weights of the parts. Had this been the case in the example just given, the pressure against each bearing would have been reduced by 50 pounds; that is, the sum of the bearing pressures would have been reduced by twice the amount of reduction of the lifting action of the belt on one pulley. This is in accordance with the conditions that are generally found in practice. Futhermore, had the lifting action of the belt been increased 50 pounds above the amount which just lifted the 200 pounds weight of the lower pulley, the pressures on each pair of bearings would also have been increased by 50 pounds. The shaft of the lower pulley would then be pressed upward against the top of its boxes.

Effect of Belt Pull with Horizontal Belt. — In a drive where the stretches of belt are horizontal and connect pulleys upon two horizontal shafts lying at the same level, the belt pull and weights of the parts do not directly add together or subtract, as in the case of vertical drives. The weights of the parts still act vertically but the pull of the belt acts horizontally, so that the two forces causing pressure upon the bearings act at right angles to each other. Therefore, if the pressure upon each pair of journals for a horizontal drive is 200 pounds and the belt pull is also 200 pounds, the resultant pressure due to these forces acting upon a pair of bearings supporting the pulley may be found graphically by drawing a horizontal line the length of which (according to some convenient scale) represents 200 pounds. The scale may be taken as one inch of length for 200 pounds. This line may be taken to represent the belt pull. A second line, drawn vertically from one end of the horizontal line and also one inch in length, may represent the 200 pounds pressure upon the bearings due to the weight of the parts. A line joining the extremities of the two just drawn and forming the hypothenuse of a right-angled triangle will represent the resultant pressure against the bearings, both in amount and direction. By measurement, its length will be found to be 1.41 inch, which is equivalent to 282 pounds. This does not take into account the weight of the belt; half of the total weight of the belt would have to be added to the vertical force. This weight depends upon the distance between the shafts as well as the cross-section of the belt. If the belt tensions are increased so as to make the horizontal force (tending to pull the shafts against the bearings) 250 pounds, the resultant pressure against the bearings will be 320 pounds.

In the actual practice of transmitting power mechanically about shops and factories, the weights of the parts are very much smaller in proportion to the belt pull than those given in the foregoing. As an example, a pressed steel pulley 30 inches in diameter with an 8-inch face weighs about 74 pounds. If its shaft weighs 76 pounds, the total weight tending to cause pressure downward upon the bearings will be 74 + 76 = 150 pounds. The pull due to the tensions in the two stretches of belt will often be as great as 1000 pounds or more. Combining this horizontal force of 1000 pounds with the vertical force of 150 pounds gives 1011 pounds as a resultant pressure against the bearings. This is only a little more than one per cent greater than the sum of the tensions in the belt. The weight of the belt, in this case, is again neglected; its effect would be correspondingly small except for unusually long stretches of heavy belt.

Effects of Belt Tensions under Centrifugal Action. — In order to clearly understand the amount of pressure that is caused by the belt tensions against the journal bearings, and also the effect of centrifugal action, a practical example will be considered. In order to simplify the matter, the weights of the pulley, shafting, and belts will be omitted, now that the nature of their effect has been pointed out. The solution is thus left in such a condition that any weight of parts can be added or accounted for.

As an example, suppose that the pressure against the two bearings supporting a pulley and its shaft is 200 pounds when there is no belt on the pulley, the bearings being on opposite sides of the pulley. If a belt is now placed on the pulley and wrapped halfway around it so that its two stretches are parallel and lead off vertically downward, the pressure against the bearings will be increased by an amount equal to the sum of the two belt tensions, $T_1 + T_2$ (total tensions on tight and slack sides of belt, in pounds), at the extremities of the arc of contact, plus the weight of the part of the belt wrapped on the pulley, *if the belt is at rest.* Thus, if the tension on one side is 400 pounds, and 600 pounds on the other, and the weight of the part of the belt on the pulley is 10 pounds, the total pressure is 200 + 400 + 600 + 10 = 1210 pounds, when the belt is at rest. If the belt is running, and the resistance to the turning of the pulley is such that the total tensions will still remain of the same values as before, the pressure on the bearings will become less than when the belt was at rest; because part of the tension is now due to centrifugal force on account of the speed and weight of the moving belt. The value of this centrifugal tension in each square inch of cross-section of the belt is obtained by the formula:

Centrifugal tension $= 0.3727\, wv^2$ pounds per square inch,

in which w = weight of one cubic inch of belt in pounds;
v = the velocity in feet per second.

If the velocity is 80 feet per second, then:

Centrifugal tension $= 0.3727 \times 0.035 \times (80)^2 = 83$ pounds per square inch of cross-section.

If the cross-section of the belt is 2 square inches, the total centrifugal tension in each stretch will be $2 \times 83 = 166$ pounds. This centrifugal tension has no effect toward drawing the belt against the pulley, and, consequently, causes no pressure on the journal bearings. The only part of the tension that is effective in turning the pulley is what remains after deducting the centrifugal tension from each of the total tensions T_1 and T_2. The values of the effective tensions are therefore:

$$600 - 166 = 434; \text{ and } 400 - 166 = 234 \text{ pounds.}$$

It is these *effective* tensions of 434 and 234 pounds which cause pressure against the bearings when the belt is running at 80 feet per second, instead of the *total* tensions of 600 and 400 pounds which caused bearing pressure when the belt was at rest. The total pressure on all the bearings when the belt runs at 80 feet per second is, therefore, 200 + 434 + 234 + 10 = 878 pounds. This is less than 73 per cent of the total bearing pressure of 1210 pounds, when the belt was at rest with the same total tensions in it.

Effect of Weather on Belts. — The pulling power of belts may vary considerably owing to different weather conditions, the amount of power often being reduced

greatly in damp weather. In carrying out some tests with a drill press, it was found impossible to obtain uniform results owing to different belt tensions. In one case, for example, using a driving pulley about 16 inches in diameter with a vertical belt to a counter pulley of about the same diameter, with an oak-tanned belt, considerably used, the results indicated that at one time the driving power of the belt was not more than two-thirds of what it was at another time; in fact, in one instance, it was only one-half. In view of the result of this experiment, where the relative power delivered could be accurately determined, there seems to be no doubt about the statement as to the pulling power varying with the weather conditions. The losses are much less with chrome-tanned leather belts than with oak-tanned belts. Another peculiar result found from these tests was that a new oak-tanned belt would pull only about 50 per cent of what an old belt would.

Tests with Rope and Belt Drives. — Some experiments were made at the Technical Institute at Charlottenburg, Germany, in order to establish definitely the loss of energy in transmitting power by means of belts and ropes. These experiments indicate that the capacity of belts for transmitting power at high velocity is considerably higher than given by ordinary formulas and calculating methods. It is an advantage in belt drives to use large pulley diameters, and consequently high peripheral velocity. Considerably better results are obtained if the pulleys are made from wood for diameters less than 24 inches. An idler for obtaining the required tension, placed close to the small pulley and having a diameter at least 50 per cent greater than this pulley, does not reduce the efficiency of the drive to any appreciable extent when the belt speed is not greater than 100 feet per second. An idler for producing proper tension and for providing larger angle of contact on the smaller pulley is, therefore, to be recommended in belt drives of high ratio between the sizes of the pulleys.

The friction in the bearings is considerably greater for rope drives than for belt drives, because ropes must be under higher tension than belts, all other conditions being equal. The efficiency of rope drives decreases considerably for high speeds, while it increases for belt drives. When high peripheral velocity is required, it is advisable to use belts rather than ropes. In the case of rope drives, the efficiency is also reduced when the diameter of the pulley or sheave is reduced, due to the power required for bending the rope, whereas the diameter of the pulley in belt drives is of very little consequence, within rather large limits. The highest efficiency obtained during the tests referred to was 98 per cent with the belt drive, all conditions being ideal. Power transmission by means of a single rope gave 97 per cent as the highest efficiency; using two ropes side by side made the highest efficiency 95 per cent; and only 90 per cent efficiency was obtained with four ropes side by side. The efficiency in the case of rope drives thus decreases about in proportion to the number of ropes employed. The figures quoted give the actual efficiency of the drive, the friction losses in the bearings being subtracted. The latter are also greater in the case of ropes than in the case of belts, due to the higher tension required in rope drives. While idlers, in the case of belt drives, did not appreciably decrease the efficiency, idler sheaves, in the case of rope drives, were found to decrease the efficiency considerably.